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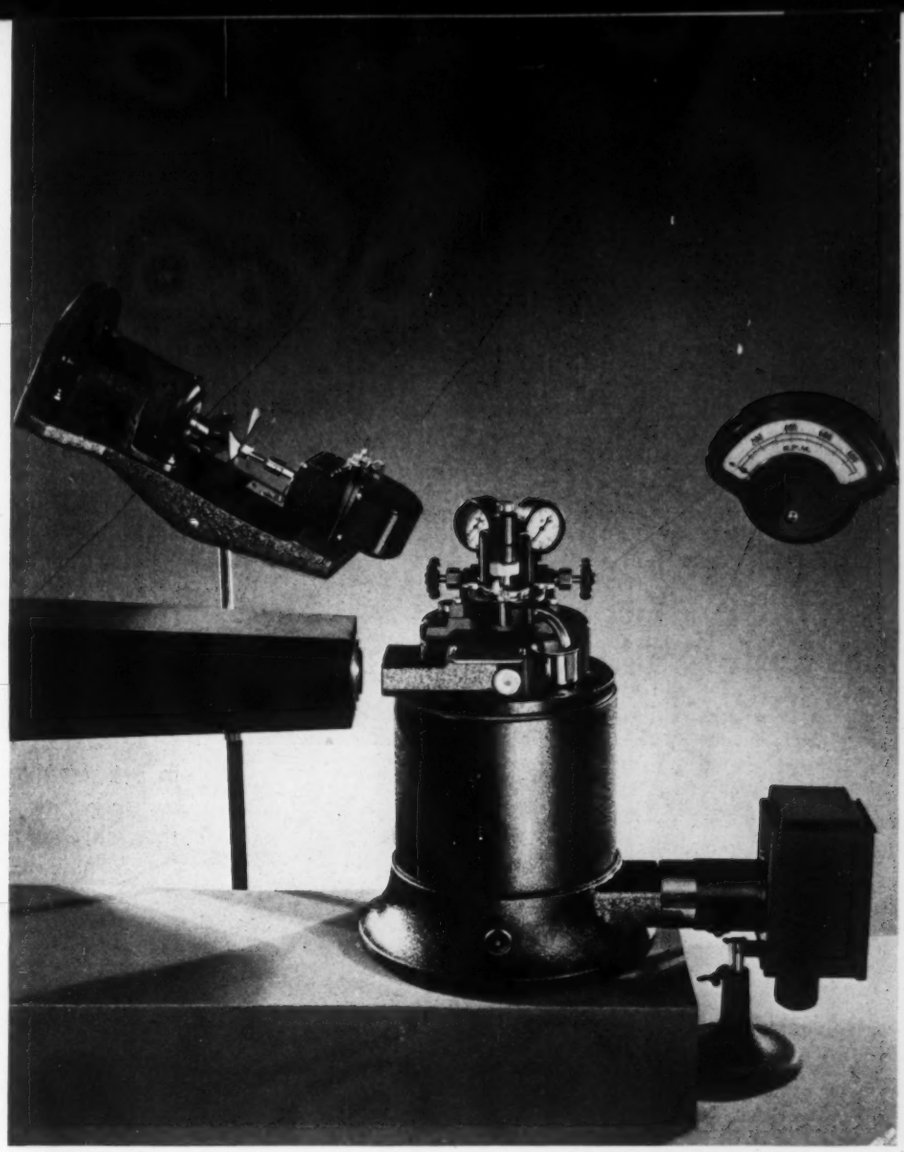


Fig. 1—Ultra-centrifuge, a machine designed for safely-controlled generation of stupendous centrifugal force, is one of the latest and most useful "tools" for the research laboratory

80,000 Revs Per Minute? Modern Design Does It!

By Arthur U. Ayres
Chief Engineer
The Sharples Specialty Co.

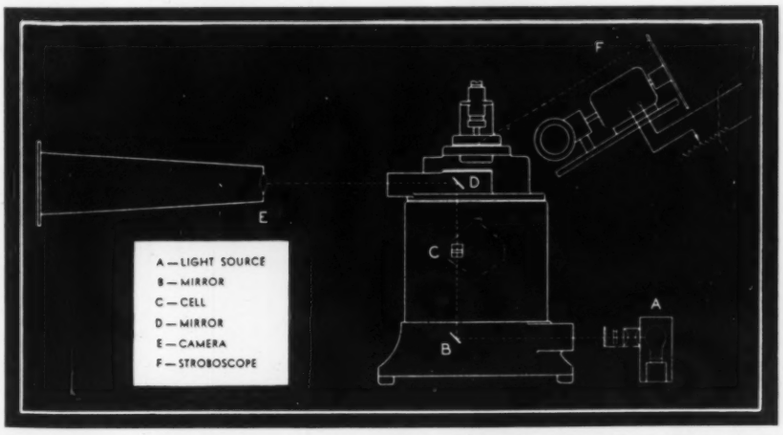


Fig. 2—Schematic diagram showing optical systems through which speed of ultra-rotor is determined by stroboscope, and through which progress of sedimentation is recorded photographically

UNIQUE design problems have been solved in recent development of Sharples ultra-centrifuge. This machine is simple and in spite of its exceedingly high operating speeds, its power consumption is low and wear over long periods is negligible. This unit has been developed to make possible the accurate determination of the molecular weight of colloids by the application of Stokes Law to a determination of sedimentation velocity, the machine being employed in a manner described in some detail later in this article.

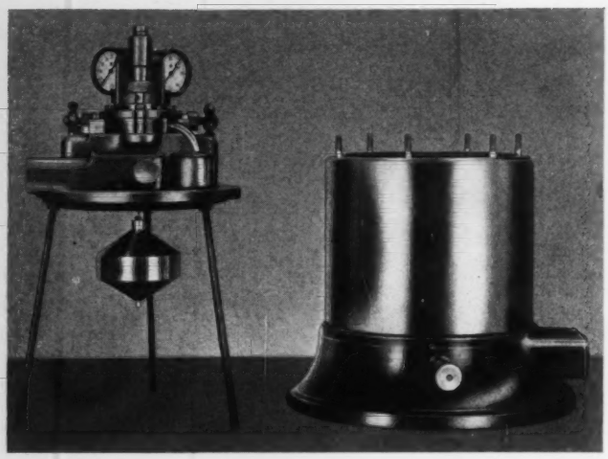


Fig. 3—Driving head removed from housing, showing ultra-rotor attached to spindle in running position

The assembled unit is shown in *Figs.* 1 and 2. Its rotor, which can be seen in *Figs.* 3 and 4, has a maximum diameter of 3½ inches and is pierced at a radius of 1¼ inches for the transparent cell containing the sample to be tested. It is made of the highest strength Duralumin forging, shaped for maximum resistance to distortion at high speed. The cell assembly consists of an aluminum ferrule in which are placed two windows of glass or quartz separated by a Bakelite disk having a sector-shaped slot in the center. The sample is introduced through an opening in the wall of the ferrule. Windows and Bakelite spacer are fastened together with Picein cement to make joints leak-proof. Assembly is tightened into rotor by a nut.

Maximum Safe Speed Is 1350 r.p.s.

To give the machine maximum of usefulness it must be operated at highest possible speed consistent with safety. The rotor just described has been operated at speeds as high as 2000 revolutions per second, representing a centrifugal force of 485,000 times gravity. This is unsafe, however, as it develops stress beyond the elastic limit of the metal. Therefore commercial operating speed has been set at 1350 revolutions per second, developing a centrifugal force of 250,000 times gravity. High speeds must be accomplished with minimum power consumption and with assurance against bearing failure. Therefore the design is built around the principle of Henriot and Huguenard based on the Bernouilli theorem, thrust being taken on a film of air, eliminating mechanical contact.

To reduce power consumption and to prevent heating from air friction the rotor is mounted in a chamber evacuated by means of a mercury or Apiezon oil diffusion pump to a pressure of approximately 0.5 microns. Means are provided to prevent air ingress that would reduce this vacuum and at the same time to supply flexible bearing mountings that permit rotor to make its center of rotation its center of mass, thus avoiding vibration.

It will be noted in *Fig.* 4 that the rotor is suspended by means of a collet from a shaft of very small diameter, with a turbine wheel, *Fig.* 5, fastened to a point near its upper end. Two bronze radial bearings are mounted beneath the turbine wheel and one is arranged above it. These are pressed into tubular housings centralized by Neoprene rings at each end. The upper bearing is adjustable in position to avoid nodes that might build up vibration. This establishes correct conditions for vibrationless running, combining an extremely flexible shaft with flexibly mounted bearings having a minimum of mass between shaft and cushion supports.

Air at approximately 5 pounds per square inch is introduced into the lubricating oil reservoir, *Fig.* 4. It serves the dual purpose of applying pressure to lubricant fed past the sleeve bearings and of creating an "air thrust bearing." The air passes through channel *A* (*Fig.* 5) into the cavity beneath the turbine wheel, lifting the wheel sufficiently to permit air escape on the full periphery past the Bakelite ring *C* on which the wheel normally rests. Note that the exposed upper face of this ring is greater in area than the lower face. This permits

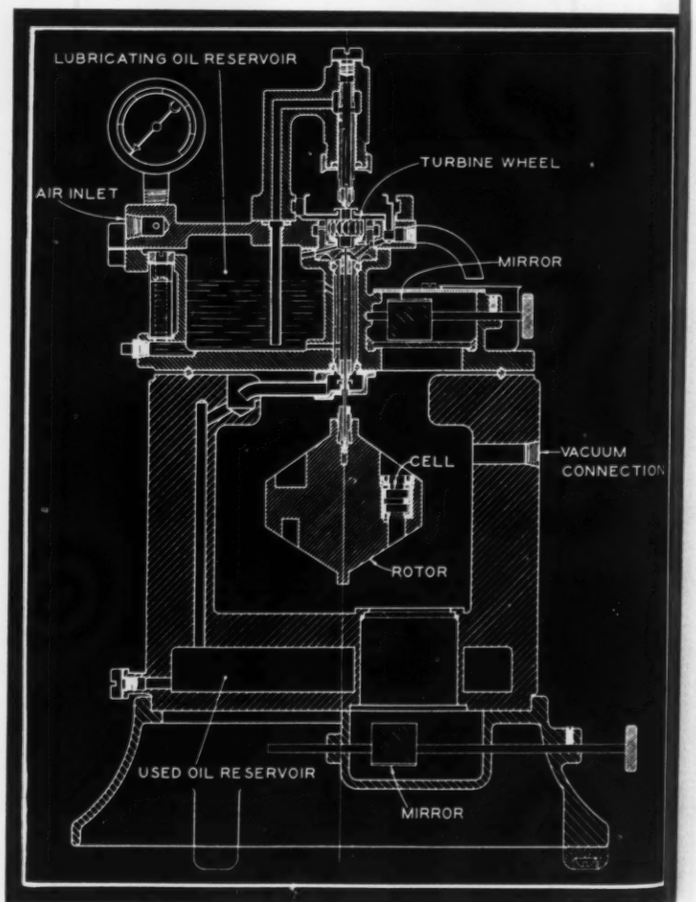


Fig. 4—Cross section of ultra-centrifuge, depicting lubricatio driving, and bearing systems, and how light passes through c

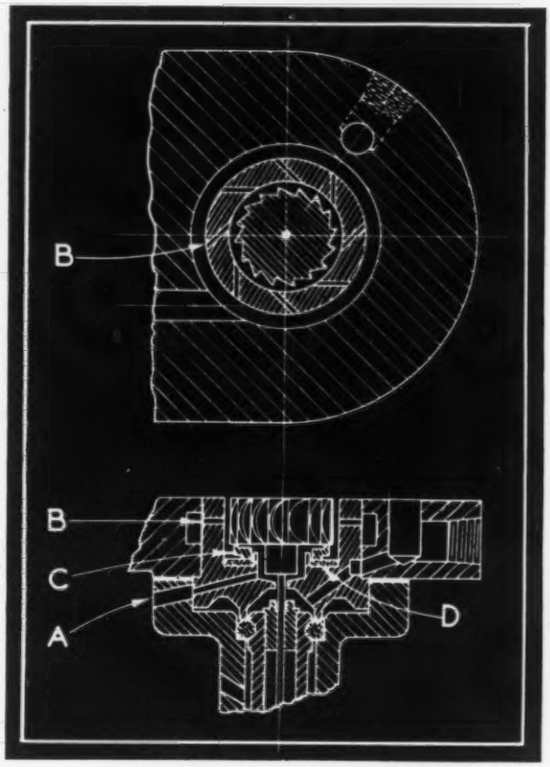


Fig. 5—Details of compressed air turbine, revealing how rotor is carried on air thrust bearing

air pressure to force the Bakelite piece firmly against the Neoprene disk *D* while lifting the turbine wheel. This wheel, together with spindle and rotor, rises approximately 1/64-inch and so remains as long as lifting air is provided. Higher pressure air fed through a channel, not shown, impinges against milled buckets of the turbine wheel through tangential channels *B*, thus causing rotation without endwise thrust.

Flexible Rings Serve Dual Purpose

The 5 pounds air pressure in the reservoir forces oil past all three sleeve bearings, waste from the upper two discharging to the outside and that from the lowest passing to a reservoir inside the vacuum chamber. The oil under pressure squeezes the Neoprene rings against extending flanges, sealing these points against oil escape. Thus these rings serve the dual function of oil seals and flexible bearing mountings. No air can enter the vacuum chamber because all available channels are occupied by high viscosity oil under pressure.

The rotor is surrounded by a nickel steel housing 1½ inches thick to protect operator should he raise the speed beyond safe limits. Windows are provided at top and bottom of the housing, in line with the cell location. Beyond these are mirrors mounted to direct a beam of light through the cell and thence to the lens of a camera, as depicted by *Figs.* 1 and 2.

Speed is determined by observing a spot on the upper oil shedder through the slotted disk stroboscope, *Figs.* 1 and 2. This consists of a motor with double shaft extension. To one end is coupled a magneto connected with an indicating voltmeter calibrated in revolutions per minute, and on the other end is mounted a disk pierced by 100 slots. Beyond the disk is a blank plate with a single observation slot. A carbon pile rheostat permits close adjustment of stroboscope motor speed, which is varied until the spot under observation appears as a stationary arc extending one-quarter way around the oil shedder. A direct reading of the motor speed multiplied by 100 gives revolutions per minute of rotor.

Procedure for Running a Test

A few drops of the sample to be examined are placed in the cell through the opening in the ferrule. This sample is then covered with a drop of such oil as is used in a vacuum pump, to prevent surface evaporation which might result in convection currents. Cell and housing are then mounted in the rotor and rotor is coupled to spindle as shown in *Fig.* 3. The assembly is now placed in its housing and the first air valve is opened until 5-pounds pressure is applied to oil chamber, thus starting lubrication and lifting rotor to operating position. The other air valve is now opened sufficiently to rotate the bowl at approximately 10,000 revolutions per minute and the vacuum pumps are started to exhaust the housing. The rotor is allowed to revolve at this low speed until the vacuum is established, which takes about an hour. This is necessary to prevent a loss of the sample through expansion of air bubbles under the surface. Higher speed tends to increase temperature of sample before vacuum is established.

When vacuum is established, air pressure is increased to between 10 or 15 pounds on the driving

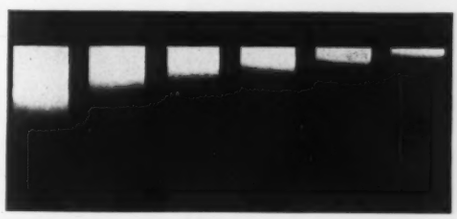


Fig. 6—Right to left—Photographs taken at intervals through sample in cell

side only and the machine is quickly brought up to operating speed of 1350 revolutions per second. The light is then turned on and the camera is mounted opposite the upper mirror, as shown in *Figs.* 1 and 2. A special plateholder provides for shifting the plate so that a number of pictures (at intervals say of fifteen minutes) can be taken on a single plate, as in *Fig.* 6, there being between plate and lens a blank baffle having one slit to limit recorded image.

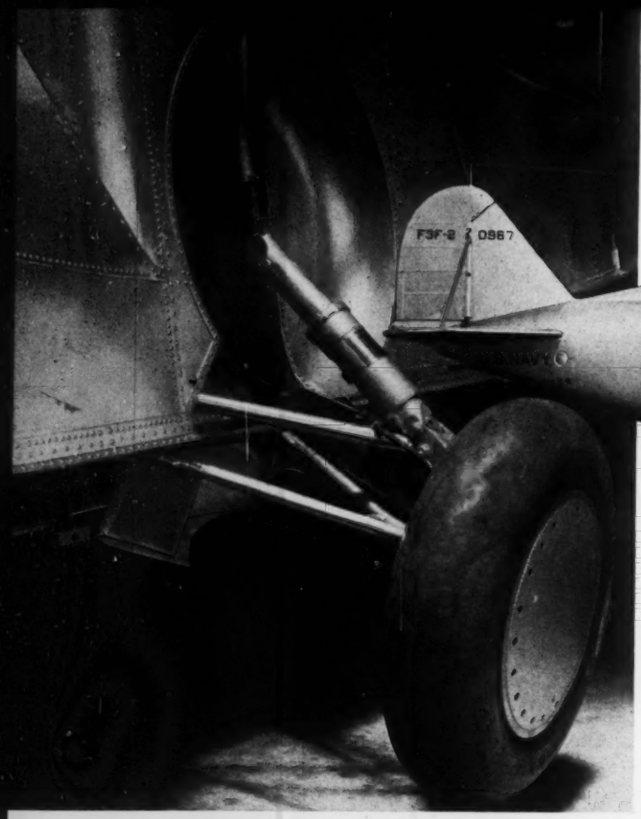


Fig. 1—Grumman navy fighter features retractable wheels which fit neatly in the side of the fuselage

Retractable Landing Gear Reflects Design

By Fred Kelly

IT IS probably no exaggeration to state that in the development of the airplane from a machine constructed primarily of bailing wire and wrapping paper in 1903 to the high-powered, all metal, highly efficient mechanism of today the greatest engineering feats of history have been accomplished. In no other machine have so many perplexing problems—seemingly insurmountable—presented themselves, and in no other machine has the finished design been so noteworthy.

One of the more recent engineering achievements in airplane design has been the refinement of retractable landing gear. Although this type of wheel mounting has been used for a number of years, the early designs were often more of an encumbrance than an asset to the machine. Today by cunning ingenuity an entire retractable landing gear assembly capable of sup-

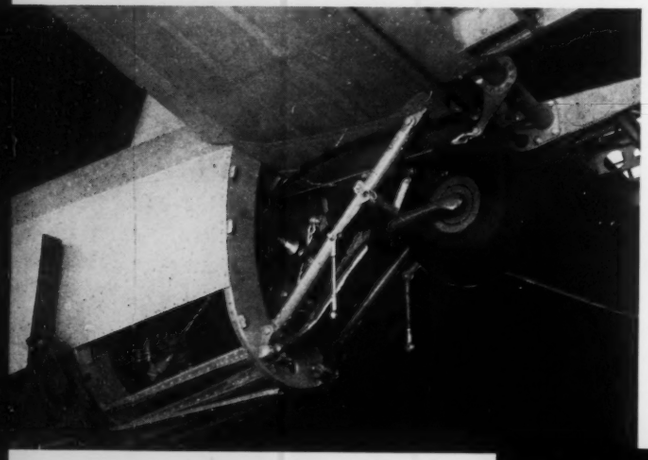
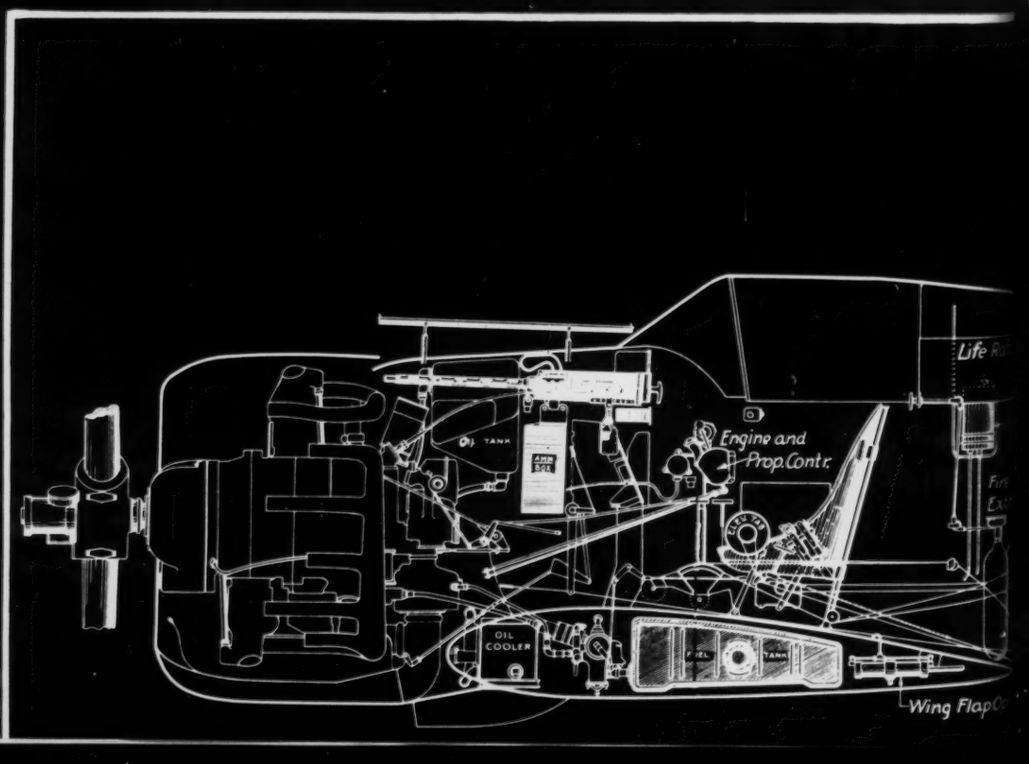
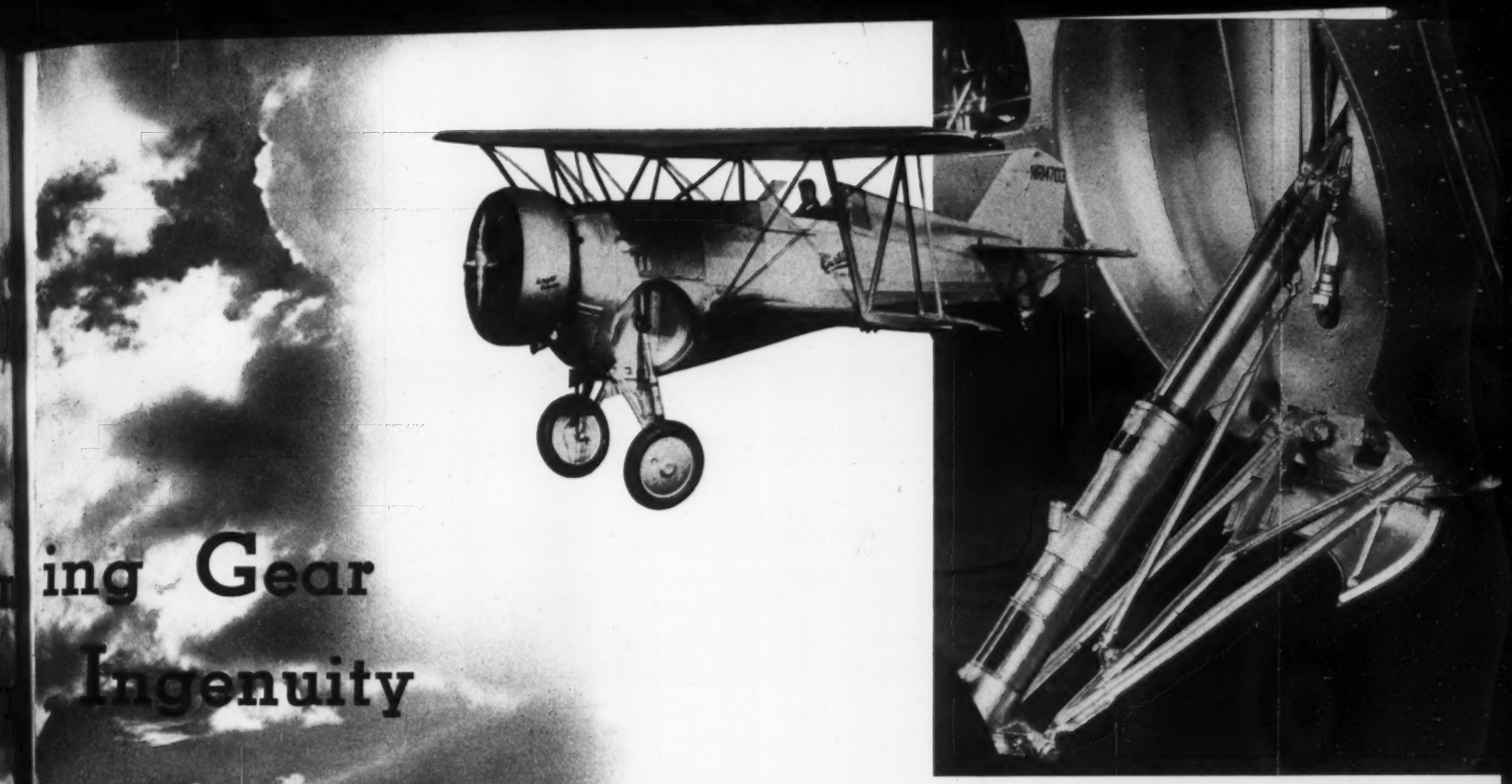


Fig. 2—Tail wheel may be retracted entirely within the fuselage of the Vought single - seat fighter by means of hydraulic control. Provision is also made for steering the tail wheel by the rudder bar





ing Gear Ingenuity

porting a load of many tons may be securely tucked away in the airplane fuselage. Reason for withdrawing landing gear into the plane body is the reduction of drag. When traveling over 150 miles an hour the additional wind resistance caused by a pair of wheels and shock absorbing apparatus extending below the airplane is sufficient to reduce speed 5 to 10 per cent, depending, of course, on the speed and type of machine.

First consideration in the design of retractable landing gear is weight. If the weight of additional mechanisms and parts necessary for folding up the wheels in the airplane fuselage offsets the gain from reduced drag there is no advantage in such a design. In the second place, unusual strength must be provided in the gear as forced landings or landings on rough terrain exert a terrific strain on the parts. Foolproof operation

Fig. 3—Long threaded screw, driven by hand crank in cockpit, operates Curtiss Hawk retracting wheels

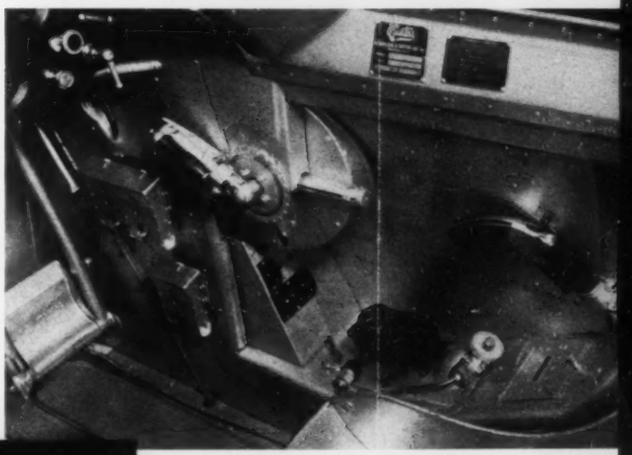


Fig. 4—Crank on right side of Curtiss Hawk cockpit enables pilot to wind up or down the landing wheels in a short time. Notches on crank quadrant lock it in position

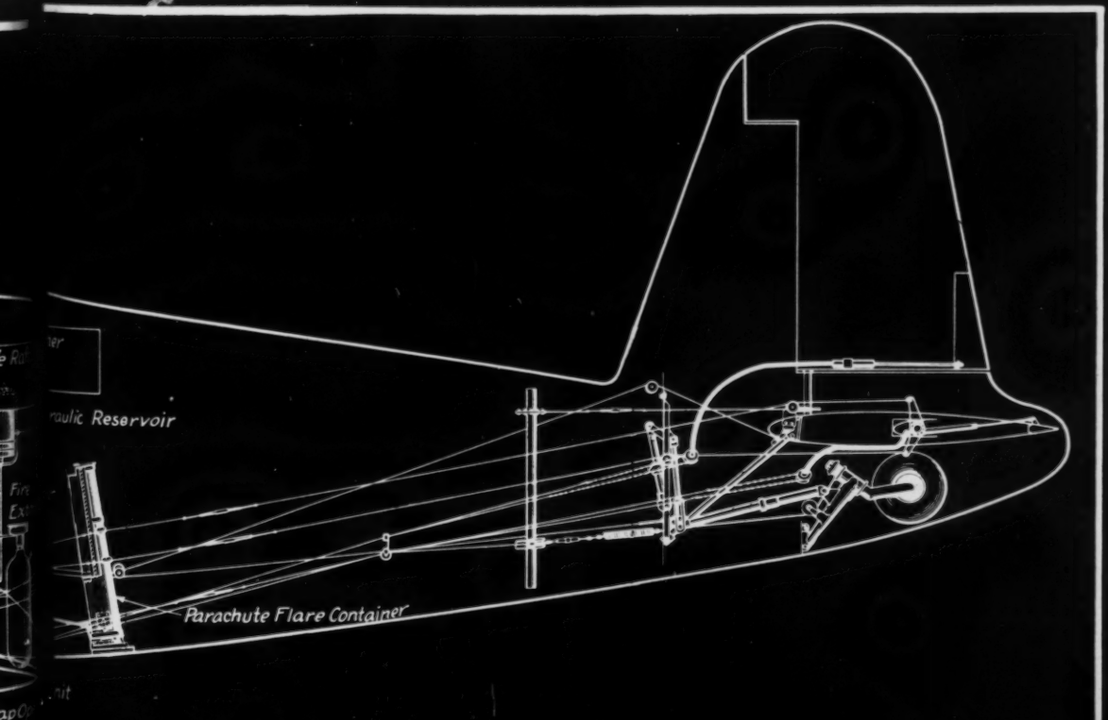


Fig. 5—Pipes extend from motor - driven hydraulic pump in front of Vought fighter to cylinder controlling retracting tail wheel in rear. Hydraulic system is also used in this airplane to control variable pitch propeller and wing flaps. Machine gun mounted in front of pilot fires through the propeller

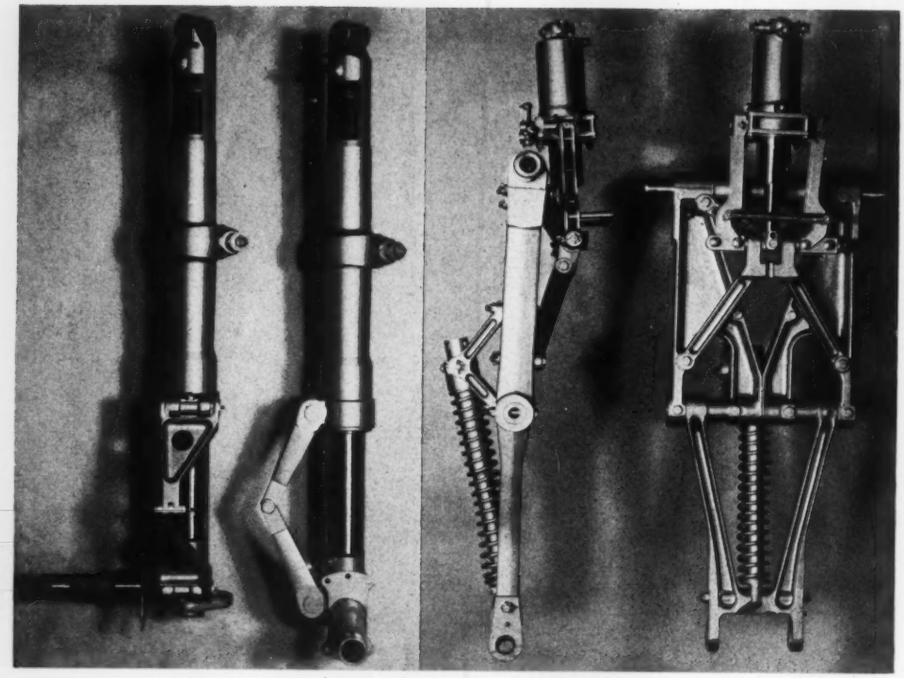


Fig. 6 — Shock absorbing struts shown in left view combine pneumatic and hydraulic snubbing action. At right is a collapsible landing gear mechanism with a telescoping hydraulic cylinder to conserve space. Spring aids in movement of parts

is the third consideration. A retractable landing gear which will not descend after being retracted is as bad as no landing gear at all.

Three principal methods are used for retractable landing gear drives: (1) Manually operated, in which the pilot supplies power by turning a crank in the cockpit; (2) direct electric drive in which a motor is connected by gears, shafts or chains to landing gear assembly, and (3) hydraulic, in which an electric motor or the airplane engine drives a hydraulic pump to supply fluid under pressure to cylinders connected to the gear assembly. In types (2) and (3) as a safety precaution, means are usually provided for manipulating the landing gear by hand in case of failure of mechanical power. Pneumatically driven retractable landing gear has been tried, but because of excessive weight, it is doubtful if any airplane made in America is so equipped.

Most popular and widely used of the three types is hydraulic. Several good reasons account for the trend to this equipment. First the fluid pressure may be obtained directly from a pump mounted on the engine or if a motor is used it can be conveniently placed in the airplane and not necessarily near the landing gear. Numerous devices such as constant speed propellers, wing flaps, de-icing apparatus and retractable tail wheels are controlled by hydraulic means and inasmuch as the system must be installed in the airplane it is used for as many functions as possible. As a safety feature, hydraulic power may be supplied from either an engine-driven pump, motor-driven pump or hand-operated pump, all on the same airplane. Usual practice is to embrace two of these systems. Other characteristics are fast operation—only six or eight

Fig. 7—At left is motor-driven hydraulic gear pump unit and at right a hand-operated plunger pump

seconds being required to lower or raise gear; quiet, smooth operation, and little chance for dirt to interfere with working parts. *Fig.* 7 shows an aeronautical motor-driven and hand-operated pump made by Pump Engineering Service Corp.

Direct electric drive is advantageous in that working parts are positively actuated and directly connected to the landing gear assembly. However, in this type a motor often is needed for each wheel on large ships and slight damage to the landing gear or the entrance of dirt in the gear mechanism will

render the system inoperative. There is a reduction in weight over a hydraulic system when it is possible to use one motor for both wheels, as piping, cylinder, piston and pump are eliminated. Particularly fast retraction and lowering of the gear is possible with a direct motor-driven system.

For small airplanes where the power required to operate the gear is not excessive, the manually-driven type cannot be excelled. This usually applies to planes with a gross weight up to 8000 pounds. As long as the pilot is conscious, he can turn the crank and exert considerable force to operate the gear assembly. Weight of landing gear mechanism is at a minimum as there are no electric motors to drive the system. Number of turns to place landing gear in position are from 20 to 40 and the pilot can turn the crank two revolutions per second. Although not as fast as hydraulic or the motor-driven type it takes only 10 to 15 seconds to lower or raise the wheels.

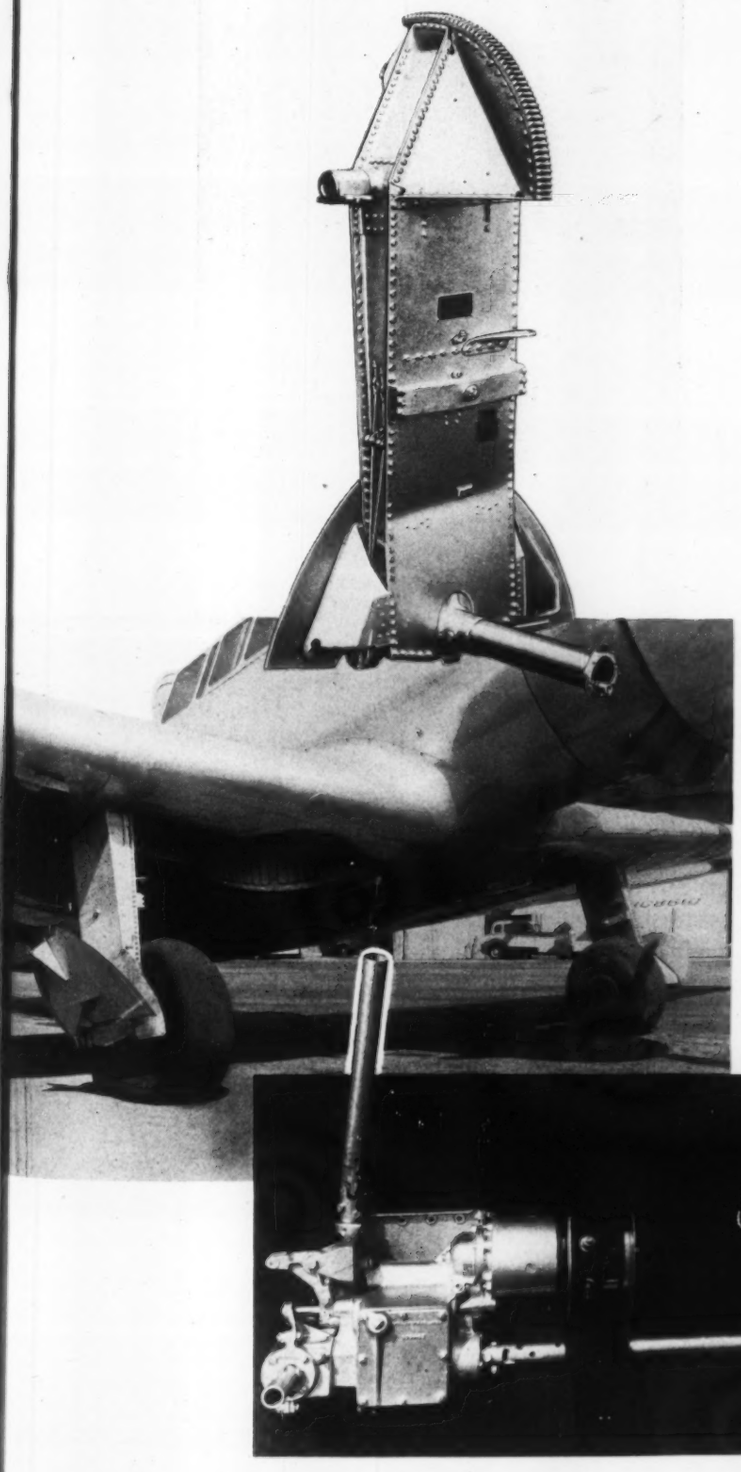
One of the problems of engineers in designing retractable landing gear is the proper cowling of the wheels after they are retracted. Unless some provision is made for blending these parts into the fuselage or wing surfaces, much of the drag remains. In addition, the landing gear must fold up in such a position that if necessary for a landing with the wheels retracted it will not interfere with the airplane sliding on its belly.

Added to these complications is the fact that an elaborate shock-absorbing and braking system is built into the assembly and must function perfectly when the wheels are extended. It is easy to comprehend the dilemma facing the aeronautical engineer in meeting all of these conditions. The system must be light in weight, yet strong and dependable. It must extend well below the airplane fuselage yet fold up so as to blend into the body of the plane. And in accomplishing these operations, the function of brakes and shock absorbing equipment must not be disturbed.

Fig. 1 depicts a Grumman navy fighter equipped with manually-controlled retractable landing gear. The close-up view shows the gear partially retracted. In *Fig.* 3 a Curtiss Hawk 111 with a similar landing gear system is shown and *Fig.* 4 gives a view of the Curtiss cockpit with landing gear crank on the right-hand side.

Retraction of the Grumman gear is accomplished by applying a torque load to the upper end of the compression strut causing it to fold inward. This rotation opens the offset hinge and puts tension in the lower

(*Concluded on Page* 116)



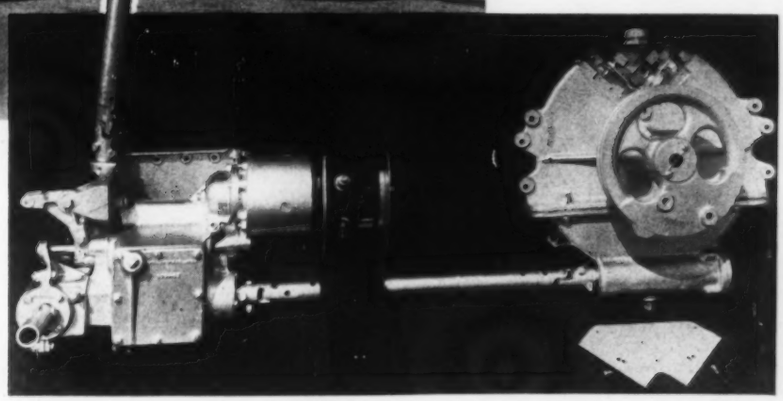


Fig. 8—Vultee bomber employs direct motor-driven retractable landing gear. Left, above, is entire strut with axle at bottom and gear segment at top. Middle view shows relation of wheel assembly to airplane, and bottom view depicts drive motor, gearbox and two shafts, one for each wheel assembly

Electric Shaver—Last Word in Compact Design

ELECTRIC dry shavers have become more than a novelty. Their fast operation, the elimination of soap and water, and relief from straight-edge and safety razors have combined to make this motor-driven mechanical device a popular as well as practical hair removing instrument. Though simple in appearance, the engineering problems that confronted the designers of these tiny units in duplicating the action of the sharp edge of the safety razor were not easily solved. Hairs must be cut as close to the face as possible without any chance of cutting or chafing the skin. Design problems were not confined to the actual shaving operation, however, for there were basic difficulties in combining a drive mechanism, electric motor and shaving head into one compact unit that would fit into the palm of the hand. Moreover, the article had to be styled to conform to critical taste.

THOUGH fast becoming a household necessity, there has been little information published on the mechanical features of electric dry shavers. The problem of designing a small instrument that would clip hairs close to the skin without chafing was a stumbling block. Only by incorporating a tiny fractional horsepower motor and unique drive in the device has it become practical

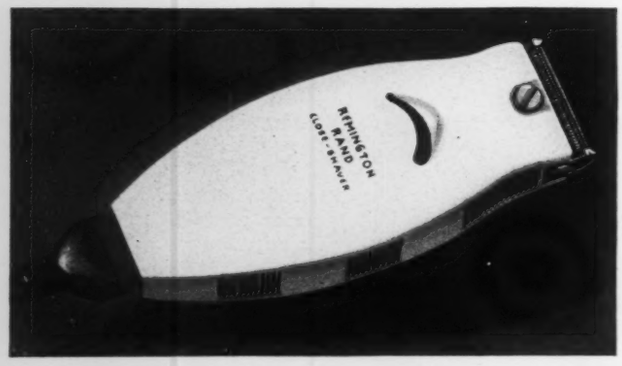


Fig. 1—Knurled wheel for starting small impulse motor is easily seen in this electric shaver model

All Shavers Fundamentally the Same

Many types of electric shavers have been developed, some encountering more success than others, but on the whole the fundamental designs resolve into the problem of cutting facial hairs in a close rapid manner. The action depends upon a comb, placed against the face, through which the wiskers extend. Cutting ensues through the shearing action of a movable lower piece, the cutter. Typical comb and cutter combinations are shown in *Fig.* 2, the movable cutter being shaded in each example. Three distinct types have been developed:

(a) Slotted combs which have been accurately ground to size

(b) Perforated combs having small, individual openings

(c) Toothed combs, somewhat similar to prevailing hair clippers.

Electric dry shavers must have a perfect fit between the reciprocating cutter and the fixed outside comb. Slight variations in spacing between the cutter and the comb would nullify the benefit of a sharp, clean cutting action. When these variations become appreciable the cutter may push the hairs to one side with-

out shearing them. Two designs are commonly employed to overcome this difficulty and insure a perfect fit of parts: (1) The cutter is pressed against the inside of the comb by spring action and (2) Parts are precision ground to close tolerances. Fortunately, in the majority of electric shavers, the wear of the inside rubbing surfaces of comb and cutter entails a "self-honing" action, resulting in a better fit with the passage of time.

The problem of designing and manufacturing a thin

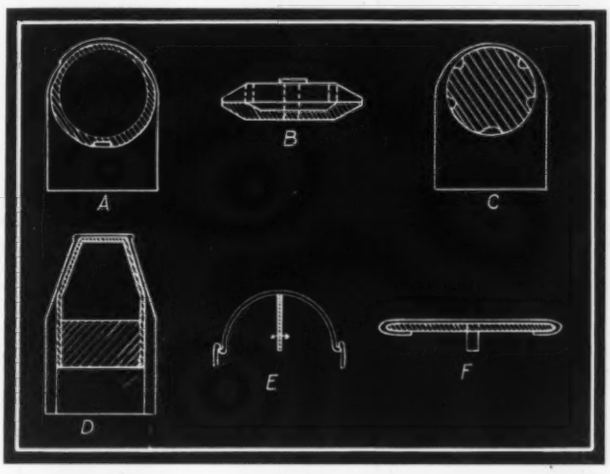


Fig. 2—Six different types of cutter arrangements in popular dry shavers are depicted

Fig. 3—Small holes through which hairs protrude are a distinguishing feature of this instrument

(about .004-inch) wall comb for picking up and admitting hairs has been difficult to handle. For electric razors employing a shuttling cutter, reinforcement of the thin sections by ribs was out of the question. The only recourse was to use a hardened steel piece. Some combs, such as *A* in *Fig.* 2, were manufactured from a solid block of metal. After slots for admitting hairs were milled into position, the opening for the cylindrical, reciprocating cutter was bored out and brought to a near-final size. The piece was then hardened and ground to final size on the inside and outside surfaces of the comb. The cutter may be prepared in much the same manner—the outside radius conforming to the inside radius of the comb. Precision grinding is required as the thin wall sections are brought down to size. There are other modifications of this procedure, whereby the metal piece may be blanked out, formed, milled, hardened and ground to final size, as was necessary for unit *D* in *Fig.* 2. Grinding is not confined to the inside and outside surfaces of the comb, for great skill must be exercised in grinding the slots to the proper size. On the average, slots for admitting hairs are about .010-inch wide—any greater width would admit too much skin and cut or chafe the face.

Small Holes Sometimes Used

There are designs which overcome this limitation of the slots by employing a large number of circular performations of .025-inch diameter. Prolonged grinding operations are eliminated in this design by the use of thin sheet metal reinforced by supporting ribs. Hairs and pores of the skin enter the circular openings, bringing the base of the hair near to, but not touching the moving cutter, a close approach to the ideal shaving operation. There are still other designs that employ the clipping action of a row of very small forked shearing blades modeled to some degree after the barber's hair-clipping machine. These toothed combs are quite narrow in cross section (see *B* and *F, Fig.* 2) and are prepared by several well known methods. They may be blanked out and formed or simply ground to size from larger pieces. However, the problem of maintaining a close rubbing action between the outside comb and the inside reciprocating cutter is still present. Care must be exercised in shaping the toothed combs to preclude any unpleasant sensation caused by rubbing the face.

A small universal, alternating or direct current impulse motor is specified for the majority of electric dry shavers on the market. There is one notable exception in which the brush type, series wound universal motor is used. The impulse motor depends for its operation upon the making and breaking of the electrical circuit delivering power to the field coil. As the field is energized a torque is exerted upon the rotor which swings around to position, but before arriving

Fig. 4—Shaver illustrated above employs cutting mechanism similar to that shown at "F" in Fig. 2

there, a cam on the shaft breaks the electrical circuit, de-energizing the field and permitting inertia of the rotor to carry it through to the next position. From a mechanical standpoint the design of these shaft cams is an important feature of the shaver motor. Laminated phenolic plastics strips are generally employed to resist wear. The shaft cams are double or triple-acting per revolution, depending upon the number of salient poles (2 or 3) in the rotor. The impulse motor

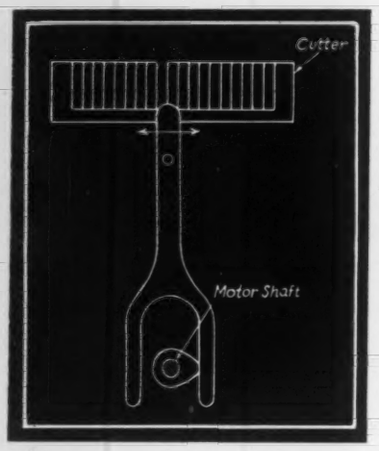


Fig. 5—Drive mechanism in typical electric shaver consists of metal stamping actuated by cam on motor shaft, which in turn moves cutter back and forth

is simple in design, being comprised of a single winding upon a laminated field structure. Electric dry shaver motors are designed to draw nine watts and operate on 115 volts, alternating or direct current. There is no starting torque, however, and a small knurled button pressed on the end of the rotor shaft is brought out through the housing for a starting twirl by the thumb.

The universal, wound series motor is more efficient from an electrical viewpoint as it has a strong starting torque and better operating efficiency. The speed of operation is slightly higher than that obtained with an impulse motor. It is noteworthy that a large amount of the motor power is directed toward overcoming friction between the reciprocating cutter and comb. When operating properly the load upon the motor in cutting hairs should have, at the most, only slight effect on the operating speed.

Drive Is Quite Simple

In fulfillment of a thorough, compact design, the drive mechanisms are made as obscure as possible. A schematic, typical drive mechanism as shown in *Fig.* 5, consists of nothing more than a sheet metal stamping, shaped to conform to the molded housing, and a pivot point near the head of the shaver. Drive is taken off the motor shaft from a cam, usually at the opposite end to that opening the contact points of the impulse motor. The driving end of the rocking arm reciprocates through a small arc to drive the cutter within the comb. Little difficulty is experienced with a simple drive mechanism of this type and there are few parts to wear or require lubrication.

Drives directly from the motor shaft, as in the unit illustrated at the top of page 24 have also been featured in recent razors. Among other prominent drive mechanisms is one employing a drive from an eccentric on the motor shaft which rocks a drive bracket through an angle of 65 degrees, giving an oscillating motion to the cutter, rather than a short reciprocating stroke obtained by other methods.

Housings Are of Molded Plastics

Small but adequate housings have been utilized for electric dry shavers. In practically every design a molded plastic housing is employed, augmenting the beauty and utility of the unit. Inserts have been molded into place, facilitating assembly of the unit. Housings may be easily taken apart for cleaning and inspection. The complete unit fits into the palm of the hand and is readily manipulated through shaving operations. Normal running of the motor during shaving hardly warms the housing and this gives no discomfort. The shaving head may be quickly removed from the housing for cleaning. Small oil holes provide entrance for lubrication to the motor shaft. The electrical plug entrance is conveniently nested and concealed in the rear, using a small plug especially suited to electric shaver design. The plastics housing reduces the chance of short-circuiting electrical parts.

From the above review it will be seen that successful electric dry shaving is contingent not only upon

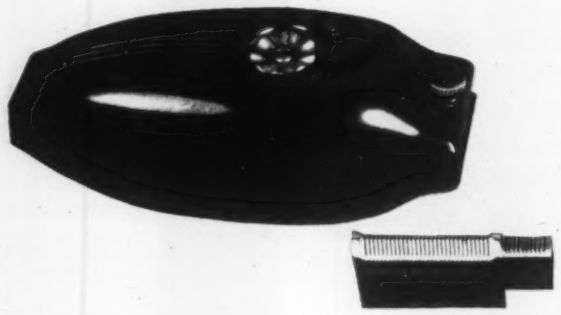


Fig. 6—Two sides to this cutter allow user to obtain either a very close or average shave

the proper design of comb and cutter with adequate motor power, but also upon accuracy of manufacture. Some electric razors are capable of giving better results than others, primarily because the design does not necessitate a manufacturing procedure that is difficult to duplicate. Good design and good manufacturing are the keynotes of success.

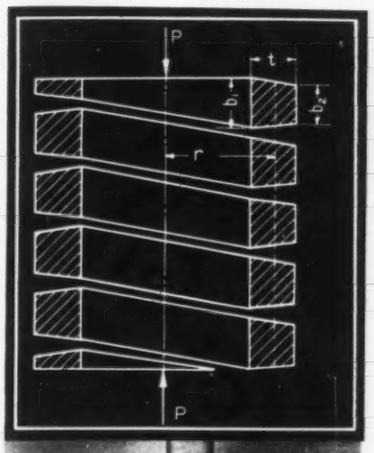


Fig. 1—Wire becomes somewhat trapezoidal due to coiling operation in a helical spring of square wire axially loaded

General Considerations in Designing Mechanical Springs

Part IV—Torsion and Spiral Springs

By A. M. Wahl

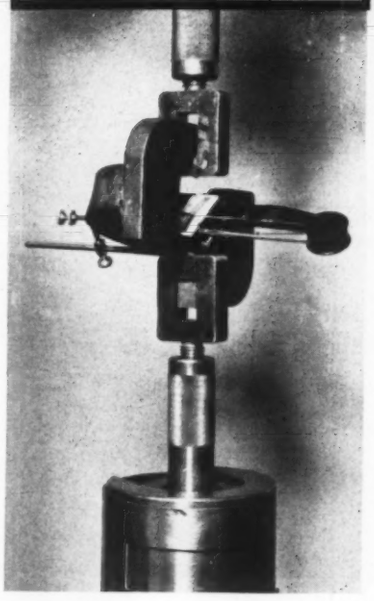


Fig. 2—Testing method is to cut a half coil from helical spring of square wire and weld two pieces to it so that semi-coil may be purely axially loaded

HELICAL springs of square or rectangular wire are frequently used in practice, one advantage of such springs being that the load deflection characteristic may be accurately adjusted by grinding off the outside of the coil. In torsion spring applications where the spring is twisted so as to put the wire in bending, a square or rectangular wire cross section makes a more efficient use of the material than does a round wire section; this is not true, however, for compression springs where the wire is in torsion.

To calculate the stress in helical compression springs of square wire, axially loaded, the following formula may be used[1]:

$$s = K' \frac{4.8\,Pr}{b^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where:

P = axial load on spring
r = mean coil radius, inches
s = max. shearing stress pounds per square inch
b = average side of square cross section, inches For the usual trapezoidal cross section (*Fig.* 1),

$$b = \frac{b_1 + b_2 + 2t}{4}$$

K' = stress multiplication factor for square wire springs; this factor depends on the ratio $c = 2r/t$ where t is the dimension shown in *Fig.* 1.

Tests made as described below indicate that a fair approximation for this factor will be obtained by using the factor K for round wire as given in Eq. 4 or *Fig.* 4 of Part II using $c = 2r/t$. A more exact expression for K' based on the theory of elasticity has been de-

THIS article concludes a series of four which have appeared in MACHINE DESIGN since the January issue. Covering the entire field of modern mechanical spring design, their value for reference is considerable and for this reason reprints of the series are being made available at nominal cost. A recognized national authority on spring design, the author is connected with the research laboratory of the Westinghouse Electric & Mfg. Co.

1. It is assumed that the pitch angle of the spring is less than 10 degrees.

rived by Gohner[2] and is:

$$K' = 1 + \frac{1.2}{c} + \frac{0.56}{c^2} + \frac{0.5}{c^3} \quad \ldots\ldots (2)$$

For an index $c = 3$ the value of K' given by this equation is about 7 per cent less than that given by Eq. 4 of Part II for round wire; for an index of 4 the difference is about 4 per cent.

To determine the curvature correction factor K_c' for square wire (corresponding to the factor K_c of Part II for round wire) the factor which takes into account direct shear is first found. Adding the stress due to the direct shear produced by the axial load to that given by Eq. 1, it may be shown that K_s' is:

$$K'_s = 1 + \frac{0.42}{c} \quad \ldots\ldots (3)$$

and hence

$$K'_c = \frac{K'}{K'_s} = \frac{K'}{1 + \frac{0.42}{c}} \quad \ldots\ldots (4)$$

where K' is given by Eq. 2.

Even in the case where $c = 3$, the difference between K_c' for square wire as given by Eq. 4 and K_c for round wire as given by Eq. 7 of Part II is only about 4 per cent and this difference decreases as the index increases. The factor K_c' may be used in Eq. 14 of Part II to determine the working stress factor C_w' for square wire springs. Since for practical springs K_c' differs from K_c by less than 4 per cent, the charts of *Figs.* 9 to 11 of Part II may be used to determine C_w' for square wire springs with sufficient accuracy for most practical work (assuming that $q = 1$ and s_y/s_c' have given values).

The charts of *Figs.* 5 and 6 Part II (which apply to round wire springs) may also be used for an approximate calculation of loads and deflections in square wire springs at given stresses. It is merely necessary to calculate the load and deflection at the given working stress in the corresponding round wire spring, i. e. one having the same outside coil diameter, number of turns, and a wire diameter equal to the average side of the square cross section. The loads thus found are multiplied by the factor 1.06 and the deflections by 0.738 to find those for the square wire spring at the given stress. For best accuracy in calculating square wire springs however, Eqs. 1 and 2 should be used rather than the charts mentioned.

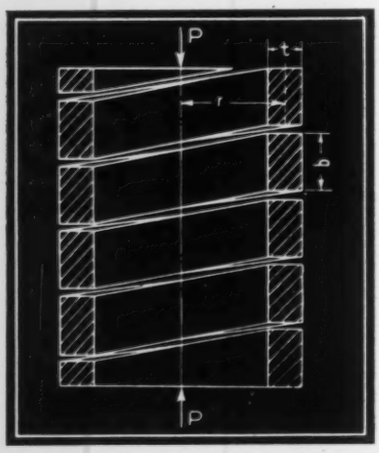


Fig. 3—Cross section of helical spring of rectangular wire axially loaded is depicted

For calculating the deflection in a square wire spring, the following formula based on St. Venant's theory is used:

$$\delta = \frac{44.6\, P\, r^3\, n}{G\, b^4} \quad \ldots\ldots (5)$$

δ = deflection of spring, inches
G = modulus of rigidity, pounds per square inch
n = number of active coils

Other symbols are as in the foregoing. A more exact calculation based on elastic theory shows that for an index $2r/t$ greater than four this equation will give values for the deflection about 2 per cent high, the error decreasing as the index increases; for an index of three the calculated value from Eq. 5 will be about 4 per cent high.

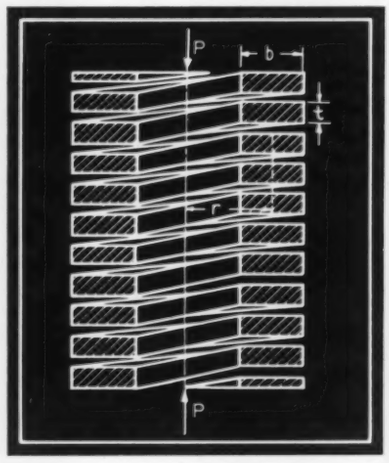


Fig. 4—Calculations are particularly complicated with helical spring coiled flatwise

For the approximate determination of loads in square wire compression springs at given deflections, the round wire spring chart of *Fig.* 7 (Part II) may be used in the following manner: First the factor C_3 is determined from this chart using the average side b of the square cross section instead of the wire diameter; the values of C_3 thus found are then multiplied by a factor 1.43. Greater accuracy will be had by the use of Eq. 5 rather than this chart, however.

STRAIN MEASUREMENTS TO VERIFY FORMULAS—In order to check the accuracy of the above formulas some tests have been made. The method was to cut out a half coil from a helical spring of square wire and weld to it two pieces to form the arrangement shown in *Fig.* 2. The eyebolts shown have spherical points so located that the semi-coil is under a purely axial load as in the case of a complete spring axially loaded. The maximum stress was measured with a special extensometer attached to the inside of the coil at the point of maximum stress. It was found that the stress

2. Zeit. V.D.I., Vol. 76, p. 271.

at this point checked almost exactly with that computed using Eqs. 1 and 2, while the stress obtained by using K factor for round wire springs was a few per cent high. In order to show that these tests on the semi-coils were representative of complete springs, strain measurements were also made on the outside of complete square wire springs loaded in compression. It was found that the stresses on the outside of the semi-coils were practically the same for a given load as the stresses on the outside of the complete springs, thus indicating that the semi-coil test arrangement does simulate the axial loading of a helical spring.

HELICAL COMPRESSION SPRINGS OF RECTANGULAR WIRE—An exact calculation of helical springs of rectangular wire is very complicated if account is taken of the effect of curvature in increasing the stress. However, the following approximate equations, based on elastic theory, will be found sufficiently accurate for most practical use. For a helical spring of rectangular wire, where the long side of the cross section is parallel to the spring axis (as in *Fig.* 3), then if $b>3t$, the maximum stress is given approximately by[3]

$$s=\frac{3}{2}\ \frac{P(2r+t)}{t^2(b-0.63t)} \qquad \ldots\ldots(6)$$

In this b, t, and r are dimensions shown in *Fig.* 3.

For cases where b/t is between 1 and 2.5, the index $2r/t>3$, and the long side is parallel to the spring axis (*Fig.* 3) the stress is given approximately by[3]

$$s=K'\frac{P\,r(3b+1.8t)}{b^2\,t^2} \qquad \ldots\ldots(7)$$

where K' is given by Eq. 2 using $c = 2r/t$. Where b/t is between 2.5 and 3, this equation will give results accurate to within a few per cent for indices greater than about 4; for indices between 3 and 4, the results may be as much as 7 per cent high.

Where the spring is wound flatwise as in *Fig.* 4 so that the long side of rectangle is perpendicular to the spring axis, the stress calculation becomes more complicated. In such cases, Eq. 7 will give values as much as 20 per cent too high, the difference being more pronounced for the smaller indices, i. e. for the smaller values of $2r/b$. (It should be noted that in the case shown in Fig. 4 the index c is $2r/b$ and not $2r/t$; hence in applying Eqs. 2 and 7 to a spring wound flatwise, the factor K' should be figured by taking $c = 2r/b$). For large indices, the values given by Eq. 7 will be good approximations[4]. In all cases the symbols b and t in this equation represent the long and short sides of the cross section respectively.

The curvature correction factor K_c'' for rectangular wire springs (which correspond to K_c of Part II) may be calculated by methods similar to those employed in Part II. Where the long side of the rectangle is parallel to the spring axis (*Fig.* 3) the method is as follows. The peak stress is first calculated using Eq. 6 or 7 depending on the ratio b/t. The nominal stress

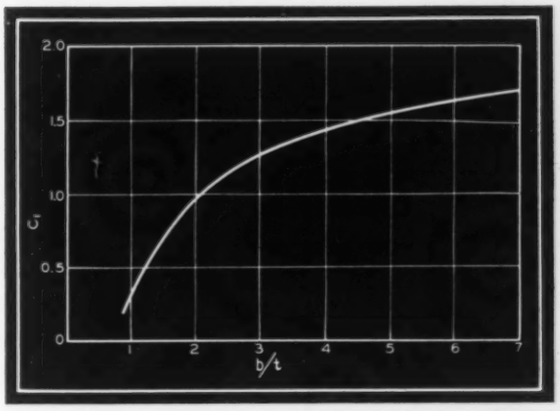


Fig. 5—Spring factor may be easily determined by taking values from this curve

is determined by adding the stress calculated from Eq. 7 taking $K' = 1$ to the stress due to direct shear assumed uniformly distributed over the cross section (the latter is P/bt). The factor K_c'' is then equal to the peak stress divided by the nominal stress. This factor may then be used instead of K_c in Eq. 14 of Part II to determine the maximum stress s_{max} or to determine the working stress factor C_w'' corresponding to the factor C_w for round wire.

Where the spring is coiled flatwise as in *Fig.* 4, determination of K_c'' also becomes very complicated, since the peak stress may be either on the long or the short side of the rectangle, depending on the ratio b/t and the index $2r/b$. For this reason it will not be discussed here.

To calculate deflections in rectangular wire springs having large indices, say greater than about 8, the following formula, based on St. Venant's theory, will give results accurate to within a few per cent:

$$\delta=\frac{19.6\,P\,r^3\,n}{G\,t^3(b-0.56t)} \qquad \ldots\ldots(8)$$

In this b represents the long side of the rectangular cross section, t the short side and r the mean coil radius (*Fig.* 3 or 4). The other symbols are the same as those for Eq. 5.

If higher accuracy is desired, for rectangular wire springs of large index, the following formulas may be used[5].

$$\delta=\frac{2\,\pi\,P\,r^3\,n}{G\,C''} \qquad \ldots\ldots(9)$$

where

$$C''=t^3\left[\frac{1}{3}b-0.209\,t\left\{tan\,h\left(\frac{\pi}{2}\frac{b}{t}\right)+0.004\right\}\right] \qquad \ldots(9a)$$

If $b/t>2.7$, the factor C'' becomes:

$$C''=t^3\left(\frac{1}{3}b-0.21t\right) \qquad \ldots\ldots(9b)$$

Where the long side of the rectangle is parallel to

3. V.D.I., 1933, p. 892.
4. For the smaller indices, if a more exact calculation is desired the charts by Liesecke, V.D.I. 1933, p. 425, may be used. It should be noted that the chart for deflection calculations on p. 425 is in error. See p. 892 of this volume (V.D.I. 1933).
5. V.D.I. 1932, p. 271. It is assumed that the pitch angle is less than 12 degrees.

the spring axis (as in *Fig.* 3) Eq. 9 will give results accurate to within a few per cent even for indices as low as 3, accuracy increasing with spring index[6].

Where the spring is coiled flatwise as in *Fig.* 4, for spring indices greater than about five, Eq. 9 will also

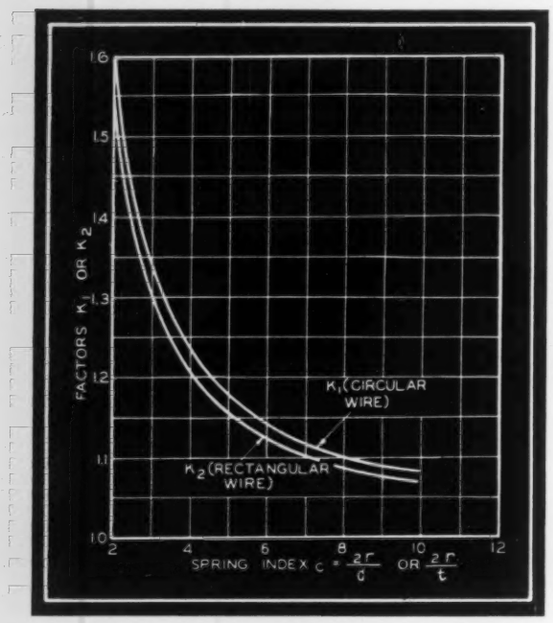


Fig. 6—Torsion stress concentration factors for circular and rectangular wire are shown

give results accurate to within a few per cent. For the smaller indices, and for the larger indices if higher accuracy is wanted, where the spring is coiled flatwise, the following equation is used[5]:

$$\delta = \frac{2\pi P r^3 n}{G C''\left(1+\frac{C_1}{c^2-1}\right)} \quad \ldots\ldots (10)$$

where C_1 is given by the curve of *Fig.* 5 as a function of b/t, $c = 2r/b$ is the spring index (*Fig.* 4), and C'' is given by Eq. (9a) or (9b). It is seen that this equation is equal to Eq. 9 divided by a factor C_2 where

$$C_2 = 1+\frac{C_1}{c^2-1} \quad \ldots\ldots (11)$$

Where the index $c = 3$ and $b/t = 4$, $C_2 = 1.18$, i. e. the results given by Eq. 9 for such cases may be around 18 per cent in error. For large values of c the factor C_2 becomes unity and Eqs. 9 and 10 become identical.

In certain instances it has been the practice in spring design to use a different value of the modulus G for springs of rectangular wire as compared with that used for circular wire springs made of the same material. Since there is no good reason why the modulus of rigidity of the material should be different for springs of the same material, whether of rectangular or circular wire, this probably has been done to compensate for inaccuracy in certain commonly used deflection formulas for rectangular wire springs. In the first series of articles under this title (MACHINE DESIGN, July 1930) it was shown that certain of these commonly used formulas were in considerable error when compared with the results calculated by St. Venant's theory (on which Eqs. 8 and 9 are based). It is the author's opinion that better results will be had if the more exact formulas such as those given by Eqs. 8, 9 or 10 are used, together with a proper value of the torsion modulus G.

HELICAL TORSION SPRINGS—Helical torsion springs are often used where it is desired to transmit torque or rotary motion. This type of spring is always loaded in such a way that the stresses set up are mainly flexural in contrast to the axially loaded helical spring where the stresses are mainly torsional. Since these springs form essentially curved bars under given bending moments, the results of curved bar theory may be used. Using this theory it was shown in the previous series[7] that the maximum bending stress for springs of circular wire is given by[8]

$$s = K_1 \frac{32\,M}{\pi d^3} \quad \ldots\ldots (12)$$

where K_1 depends on the index $c = 2r/t$ and is given by the upper curve of *Fig.* 6. In this formula, M is the applied bending moment and d the wire diameter.

Similarly for a rectangular section the maximum stress is given by

$$s = K_2 \frac{6\,M}{b\,t^2} \quad \ldots\ldots (13)$$

In this b is the side of the rectangular cross section parallel to the helix axis and t the thickness perpendicular thereto. The factor K_2 may be taken from the lower curve of *Fig.* 6 using $c = 2r/t$.

The factors K_1 and K_2 may be considered as stress concentration factors and hence may be used in a manner similar to that in which the factor K_c was used previously for figuring round wire springs (See Part II of this series).

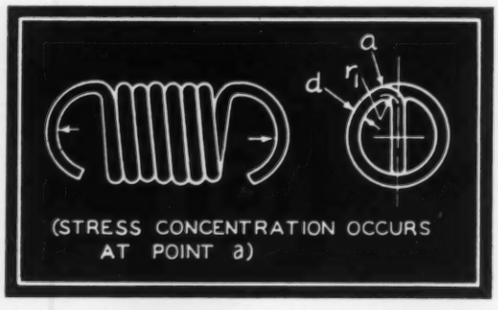


Fig. 7—Combination of bending and torsion stresses exists at "a" in helical tension spring

To evaluate the working stress s_{max} for cases where the spring is under a stress range from s_{min} to s_{max} in fatigue (these stresses being figured from Eq. 12 or 13) a similar procedure may be carried out as was

(*Concluded on Page* 135)

6. For greater accuracy in such cases the charts given in V.D.I., 1933, p. 892, may be used.
7. MACHINE DESIGN, August, 1930.
8. Note that this is a bending, not a torsion, stress.

Drive and Control Equipment Is Common to All Machines

EVERY type of machine has its own major design problem varying from high production capacity to lightness in weight. Often combinations of these or other factors are involved. Yet one problem remains that is common to all—the selection of the most suitable drive and control for the particular conditions to be met.

Questions of cost—as with the choice of any other part or material for a machine—enter largely into the specification of drives and controls. It is not necessarily the most expensive equipment that is the best—good design does not always mean high-priced design. On the other hand, there are innumerable instances where the highest possible accuracy and the finest materials are warranted, with cost a minor consideration. In both cases the primary factor for the designer to keep in mind is customer satisfaction, with resultant increased prestige to the machinery builder and a steady recurrence of sales.

To help the engineer responsible for design in solving this universal problem, to record for him some of the most up-to-date methods that are being adopted, and to present him with a work of filable reference value, a supplement on "Machine Drives and Controls" is included with this issue of MACHINE DESIGN.

As in the case of previous supplements and directories the current supplement has been stitched separately before being stitched into the center of the magazine proper. It can be removed readily for filing by unhooking two staples, without damage either to the supplement or to the magazine itself.

Scanning THE FIELD FOR Ideas

ENDLESS flat belt drive is a feature of the internal grinding attachment developed recently by Cincinnati Milling Machine and Cincinnati Grinders Inc., for their 12 and 16-inch universal grinders. This attachment, which is permanently fixed to the machine, is shown in working position in *Fig.* 1, the small view illustrating how it is swung up out of the way when main grinding wheel only is in action.

It is driven by its own 2-horsepower motor mounted on a cover plate on top of the main wheel head unit. The housing carrying the spindle is supported at the front of this plate by an adjustable antifriction bearing hinge. A single hinged bolt clamps the spindle housing firmly in working position against a machined pad and when not in use, it is held up by a powerful tension spring.

Note that the endless flat belt can be left on the pulleys when housing is swung up, making it possible to get the spindle into action quickly. Spindle speeds from 10,000 to 18,000 revolutions per minute can be maintained successfully, speed depending of course on nature of material and size of hole to be ground. In order most effectively to grind holes ranging from 5/16 to 4 inches in diameter and from 1½ to 6½ inches in depth, seven different interchangeable anti-friction bearing spindles, using abrasive wheels from ¼ to 2 inches in diameter, are available.

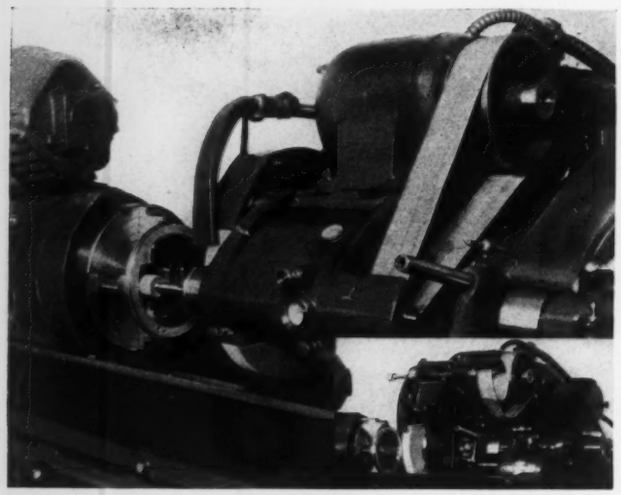


Fig. 1—Spindle of internal grinding attachment attains 18,000 r.p.m. through endless flat belt drive

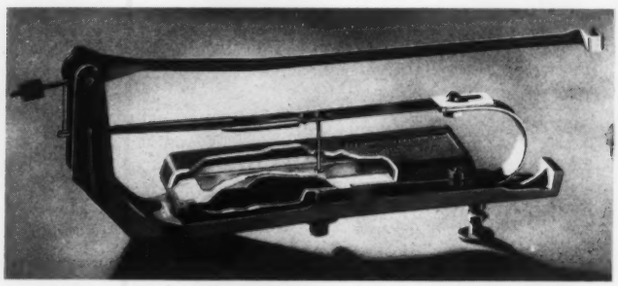


Fig. 2—Slack diaphragm of tough, durable "gold beater's skin" gives accurate draft gaging over wide range

This attachment is an excellent example of what an alert machine designer can do toward simplifying, cutting cost, and improving quality of production work. More and more parts require both internal and external grinding to exact concentricity—difficult when rechucking is involved. This attachment does away with this secondary chucking in many cases.

Leather Diaphragm Gages Draft

HOW ancient and modern materials sometimes work together to accomplish the best results in the latest mechanisms, is brought out in the dry-type draft gage developed by The Hays Corp., the design of which is made clear by *Fig.* 2.

The slack diaphragm, which is the heart of this gage and which can be seen in its die-cast housing, is of several thicknesses of tough, durable, gas-tight, colon leather, practically identical with the "gold beater's skin" of ancient times. The cantilever spring to which the diaphragm connects and through which its movements, due to changes in the force of draft, are magnified and transmitted by a length of gold bead chain to the counter-weighted pointer for further magnification, is of heat treated Beryllium copper—one of the latest engineering materials. The bearings of the pointer are hardened stainless steel pivots supported (on the taper, not on the point) in sockets in heat treated screws, also of Beryllium copper.

Economical use of these alloys for hand tools emphasizes their advantages for power equipment parts

Stresses and shocks take their toll even of hand tools. Cutting edges are dulled. Wrenches, pliers, hammers and such tools are damaged. Handles and shanks are broken as a consequence, so industry today is turning more and more to the use of Nickel alloy steels.

Light in weight, hand tools of Nickel alloy steel possess toughness, rugged strength, and excellent heat-treating qualities. They outlast hand tools of other materials many times over. They are more economical to use and maintain, and lower in ultimate cost.

The forces that destroy hand tools also shorten the life of power-operated equipment. Multiplied a hundred-fold, wear, high stresses and heavy shocks are constantly imposed on moving parts. The need for metals that will withstand this punishment is therefore proportionately greater.

The Nickel alloy steels possess in high degree the qualities required. Economical to use for hand tools, they also assure for machinery parts durability and longer life. Consultation on problems involving their application is invited.

1. These wrench sockets, manufactured by Plomb Tool Company, Los Angeles, are made of Nickel-molybdenum steel.

2. Dinging hammers and small ball pean hammers of various types, manufactured by Plomb Tool Company, Los Angeles, are made from Nickel alloy steels.

3. Bridgeport Hardware Company wrenches of S.A.E. 4650 Nickel-molybdenum steel.

4. Nickel alloy steel heavy duty screw driver made by Stanley Tools, New Britain, Conn.

5. Various types of heavy-duty pliers and pincers, manufactured by Wm. Schollhorn Company, New Haven, Conn., from Nickel-chromium steel.

THE INTERNATIONAL NICKEL COMPANY, INC., NEW YORK, N. Y.

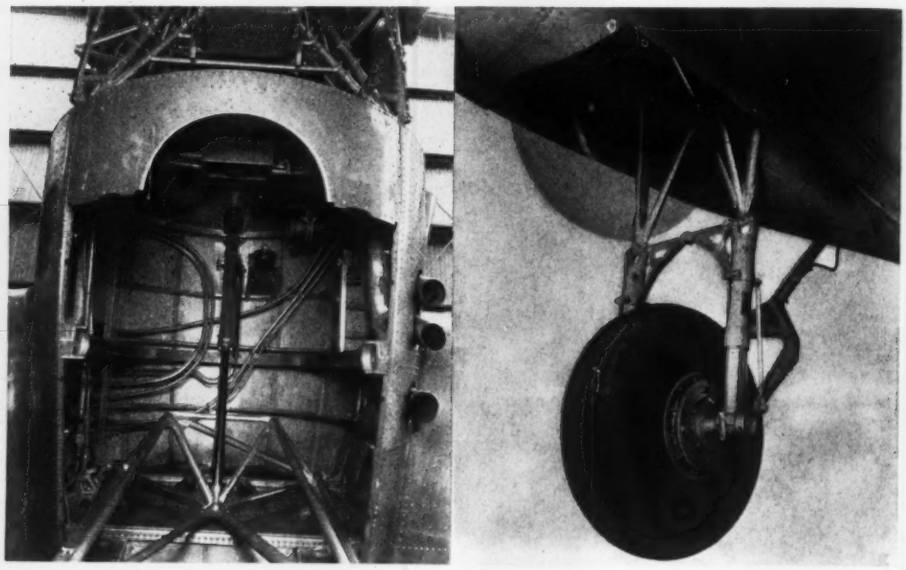


Fig. 9—Long hydraulic cylinder on Douglas transport is seen in left view, looking up into the wheel space, just behind one motor. Retracting arrangement on Douglas is shown at right

(*Continued from Page* 23)

half of the compression strut which in turn pulls the gear up and in. To reduce the retraction forces a counterbalance spring unit is connected across the offset hinge joint. This unit also serves as a lock for the offset hinge joint when the landing gear is extended in flight preparatory to landing. Two types of landing gear operating mechanisms are used: A chain and sprocket system for landplanes and a torque tube and gear box for amphibians. This latter system with its oil-contained gear boxes is resistant to the corrosive influences encountered in salt water operation.

Retracting the airplane tail wheel is a rather recent innovation and in *Figs.* 2 and 5 we see a Vought model V-143 single seat fighter with a hydraulically-controlled system for main landing gear assembly as well as tail wheel. The blueprint of *Fig.* 5 illustrates the complete enclosure of the wheel in the airplane. A special covering slips into place when the wheel is retracted leaving a smooth surface with little or no disturbance to air flow. A large oleo strut which takes the landing shock may be seen in the blueprint.

Forgings Generally Used for Parts

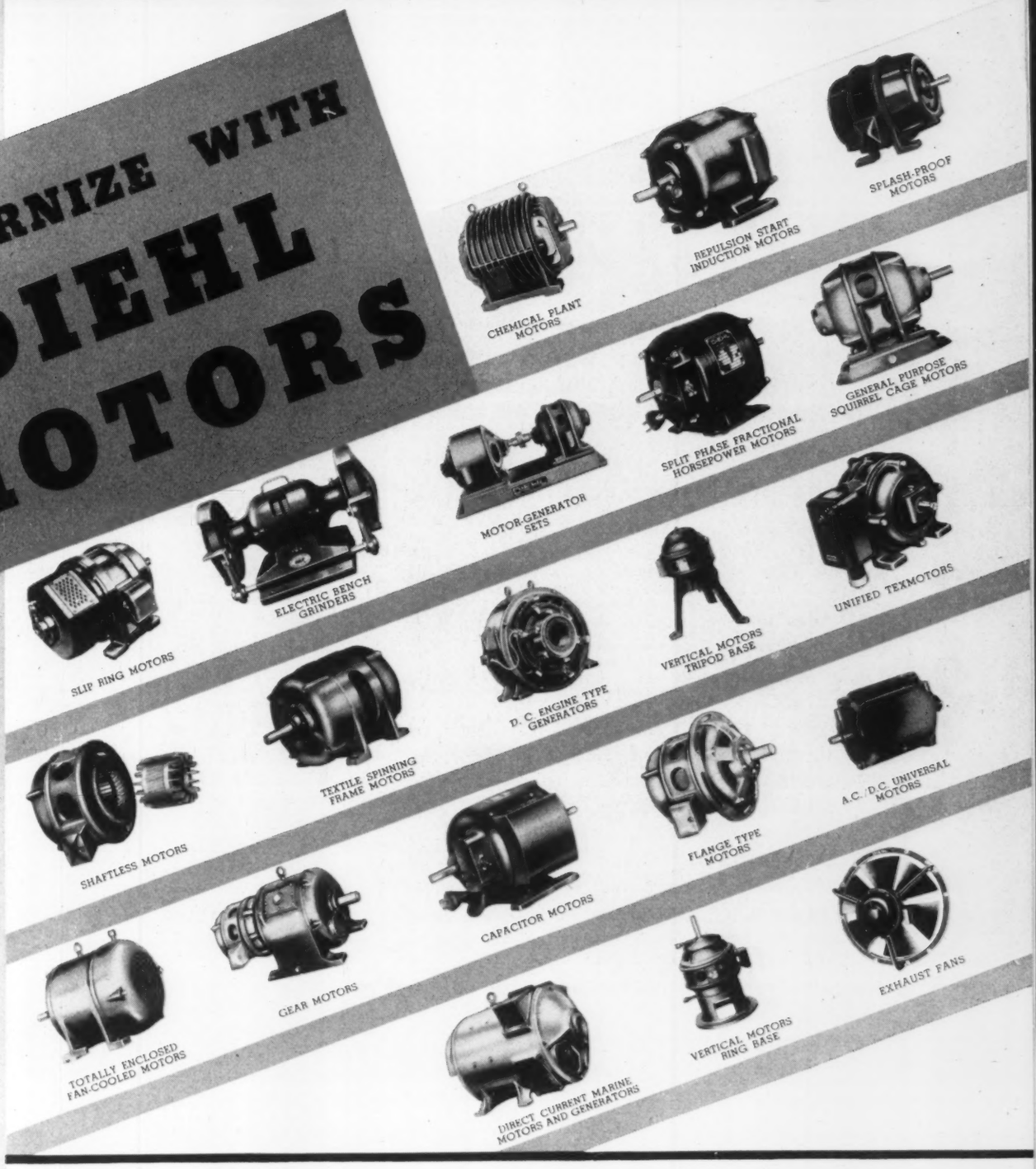
Forgings are generally used for retracting landing gear parts and *Fig.* 6 depicts the collapsing mechanism as well as two oleo struts at the left made by the Cleveland Pneumatic Tool Co. for the Vought scout model. A telescoping piston seen at the top of the right illustration folds up the mechanism with a travel of only a few inches. Assisting in this function is the powerful spring which serves to counterbalance the weight of wheels and gear. Forged steel is used for these parts and adequately supports the plane weighing about 5000 pounds. For the shock-absorbing strut the cylinder is made of a chrome-molybdenum forging and the piston is chrome-molybdenum tubing ground to precise limits.

A ½ horsepower, 12 volt motor operates the direct electric-driven landing gear shown in the three cuts of *Fig.* 8, used on the Vultee Attack Bomber, model V-11GB. The landing gear is of the full retracting type and shock absorption is obtained by oleo cylinders, rubber taxiing pads and air wheels. The landing gear consists of a strut hinged to and extending down from each outer bulkhead of the wing center section. The strut consists of an aluminum alloy box construction enclosing the oleo unit and taxiing pads. At the upper end of the strut is mounted a segment of worm gear by which the assembly is rotated into the wing. Hinged to the lower end of the strut on the outboard side is an automatically operated fairing flap, completely closing the opening in the center wing section when the landing gear is retracted.

Douglas transport employs the hydraulic system shown in *Fig.* 9. There is one hydraulic cylinder with about two-foot travel for each retractable wheel as shown in the top views. The cylinder is easily distinguished in the left view. Motor-driven pump supplies fluid power for cylinders. A double-acting ball check, plunger pump, hand operated, is also provided for possible emergency. One feature of this particular construction is that wheels, when fully retracted, protrude slightly to keep airplane undersurfaces from touching ground should a landing be necessary before the wheels could be lowered. Although propellers and other parts might be damaged the entire fuselage and wing surfaces would not be dragged on the ground.

Design developments in the aircraft industry appear so quickly that it is likely these retractable landing gear designs will be outmoded in a few years. But it is hoped that the current manner of designing these parts may offer some light on similar engineering problems. Unparalleled ingenuity is evident in each type and it is from these that further engineering advances must evolve.

MODERNIZE WITH DIEHL MOTORS



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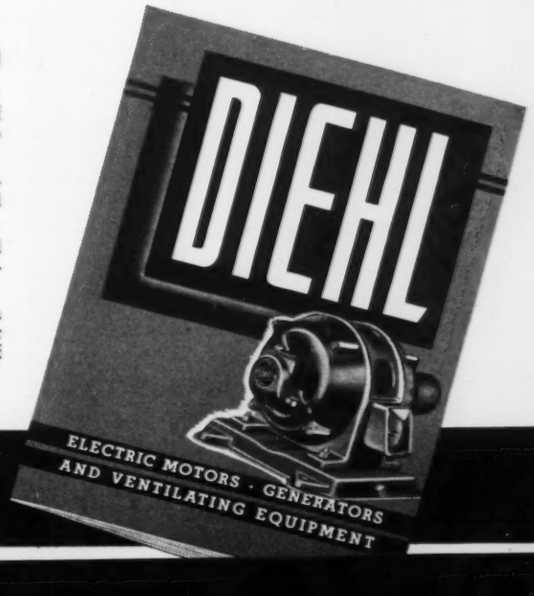
4. Because—they represent half a century of experience in motor building, and are made by a company noted for incorporating in its products approved modern features of design and construction.

WRITE FOR BULLETIN NO. 1881.



DIEHL MANUFACTURING CO., Elizabethport, N. J.
Pioneer Manufacturers of Motors, Generators and Fans
Electrical Division of THE SINGER MANUFACTURING CO.
ATLANTA BOSTON CHICAGO DALLAS NEW YORK PHILADELPHIA

Professional Viewpoints

MACHINE DESIGN WELCOMES LETTERS SUITABLE FOR PUBLICATION

Spring Washers Very Useful

To the Editor:

RELATIVE to the article on fastenings in the March issue, may I offer the following comments covering some of the advantages resulting from the specification of spring washers in design?

It is common practice, in designing such things as cylinder head studs, to use as long a stud as possible to get greater elasticity in the stud. This keeps the heads tight. It is a common principle of design that longer studs will maintain a tight joint better than short studs. This same effect can be attained by using spring or reactive washers, because they have the same physical characteristics as long studs or bolts.

We had a case where a ¾-inch bolt was holding two 3-inch angles to a plate, each of the members being ¼-inch thick. These members were subject to continual shock and it was impossible under such conditions to keep the bolts tight. This looseness resulted in annoyance and breakage. It was suggested by one of our field men that we put the heaviest reactive washer obtainable under the seat of the nut on this short bolt. The result was that we eliminated all looseness and trouble, thus preventing damage not only to the affected parts but to the adjacent ones.

Cutter Head Held Tightly

We have, on our tie adzing machine, heavy cutter heads revolving at high speed, and they must be changed every 20 to 30 minutes. If they should become loose and fly off they would be very dangerous to the workmen because of their weight. At the same time, it is necessary that the cutter heads be held on by one center bolt, and that they can be removed quickly by means of one nut. We solved this problem by using a heavy reactive washer under the nut and in all of the tens of thousands of times these heads have been put on and taken off we have never had an instance of them becoming loose.

Another instance in which the reactive washer has solved a troublesome point is on one of our machines where a heavy part is fastened to a shaft by a tapered fit and where the driven part was subject to terrific shock. We had continuous trouble with the hub loosening and with galling of the shaft. We solved this by putting on a heavy reactive washer which kept a constant pressure pushing the tapered hub on to the tapered shaft and, even though there was a slight enlargement of the hub, the reactive spring kept pushing the hub farther and farther on the tapered shaft, thus maintaining a tight fit.

Our experiences have taught us that practically any problem of looseness can be solved by using a heavier compensating device, and that spring washers—because of their greater range of action—give satisfactory results.

—H. H. TALBOYS
Nordberg Mfg. Co.

Definitely Opposed to Patent Bill

To the Editor:

I HAVE just had an opportunity to read the editorial which appeared in the March issue of MACHINE DESIGN regarding the bill now before the House Patent Committee to provide for compulsory licensing of patents.

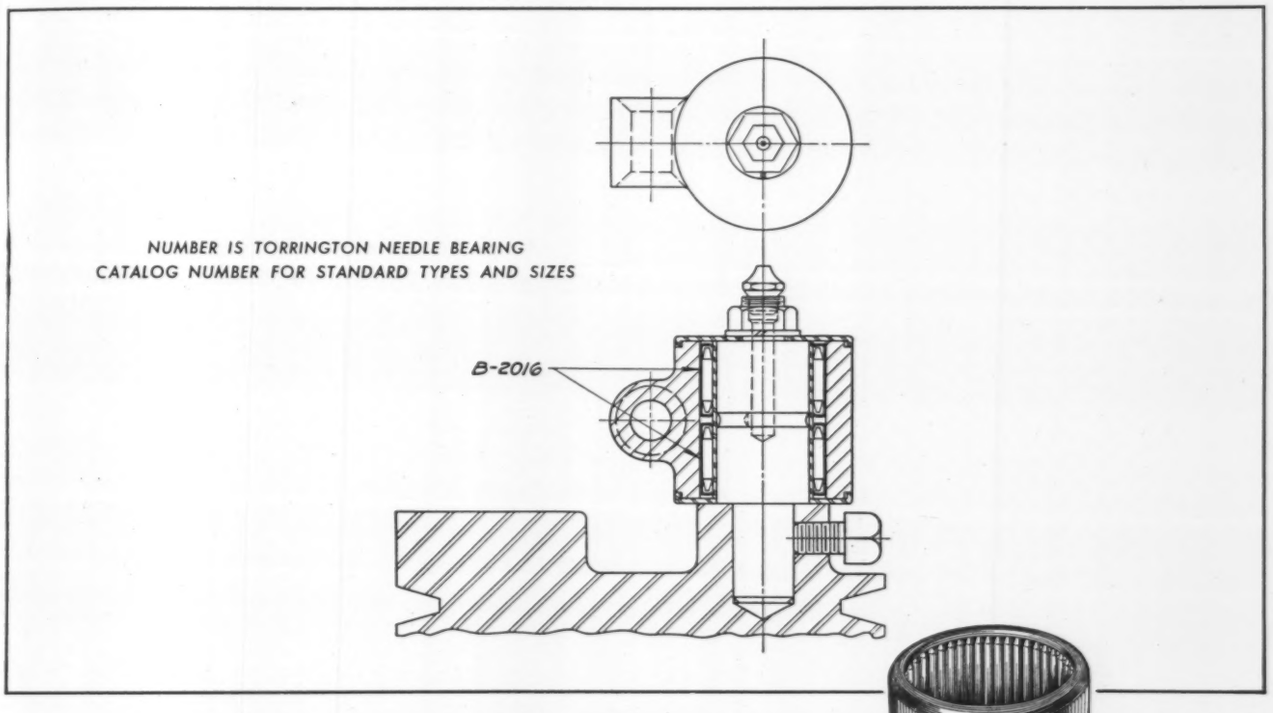
You certainly take a middle ground stand in stating that this bill has doubtful value. The bill is extremely detrimental, in my opinion, to the welfare of all machine designers. If such a bill is passed, then those companies which are well-financed can take away the benefits from research which other less fortunate companies have made. Therefore, it will be of less advantage to any company to do creative development work. Machine designers will then become "copy-cat" in nature rather than creative. Designers will also be paid considerably less because a poor man can copy that which another man of higher mental standing creates.

It is absolutely necessary for the benefit of machine designers as a class that this bill be killed in its entirety. I would, therefore, suggest that you start a campaign immediately which will remove this threat from American progress.

—ARTHUR J. STOCK
Pres., Stock Engrg. Co.

TORRINGTON NEEDLE BEARING

DESIGN AND SERVICE FEATURES



BEARING DESIGN PROVIDES EASE OF LUBRICATION

Holds Ample Grease for Pitman

Ease of lubrication of the new Torrington Needle Bearing is an important advantage in severe applications such as the one illustrated here.

The turned-in lips of the retaining shell hold an ample supply of lubricant, permitting operation for long periods without service attention. In the pitman box of a mowing machine the strain on the bearing is severe, and constant lubrication is needed. In the pitman application illustrated, a grease fitting is provided for renewal of the lubricant.

Easily Installed

Ease of installation is another marked advantage of the Needle Bearing. It is readily pressed into place in the bearing housing—an important feature in pitman boxes, which must be designed for ease of renewal.

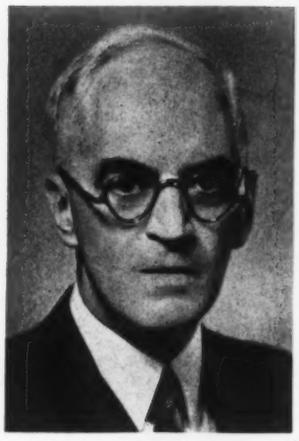
Because of the simplicity of its design the Needle Bearing is low in unit cost, making it adaptable to applications in which long service life must be combined with economical production. Moreover, the shape of the bearing—long in proportion to its diameter—permits the use of a simple housing design—a further aid to economy.

The Torrington Engineering Department offers to manufacturers the benefit of its long experience in the laying out of bearing applications, and will be glad to work with those interested in utilizing its advantages in the design of their own products. Further information is given in the Torrington Needle Bearing Catalog, available on request. Write for Catalog No. 9.

The Torrington Company
ESTABLISHED 1866
Torrington, Conn., U.S.A.
Makers of Ball and Needle Bearings
Branch Offices in all Principal Cities

TORRINGTON
NEEDLE BEARING

Men of Machines



JOHN C. PARKER

NOMINATION of John C. Parker, vice president, Consolidated Edison Co. of New York, Inc., as president of the American Institute of Electrical Engineers, brings recognition to his engineering work, characterized not only by skillful treatment of design, construction and operating problems, but also by careful attention to economic results.

After graduating from University of Michigan with M. E., M. A. and E. E. degrees, Mr. Parker spent eight years teaching, seven of these at University of Michigan when he developed one of the leading electrical engineering departments. In 1922 he joined the Brooklyn Edison Co. as electrical engineer. Four years later he became vice president in charge of engineering, and after six years, president. On reorganization of the Consolidated systems, in 1936, he became vice president of the parent company.

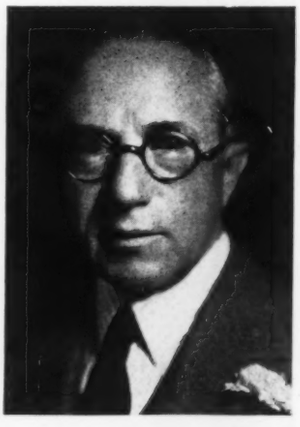
• • •

MARSHALL POST

MARSHALL POST, since 1925 vice president and works manager of Birdsboro Steel Foundry & Machine Co., Birdsboro, Pa., is the nominee for president of the American Foundrymen's association.

Mr. Post began his extended foundry career with a three-year apprenticeship as molder with the American Steel Foundries, Granite City, Ill. At the completion of this course he joined the Commonwealth Steel Co. as journeyman molder. Later he returned to the American Steel Foundries as assistant foreman, finally becoming foundry superintendent. He accepted a position as superintendent of foundries in 1918 with Marion Steam Shovel Co., but in two years again returned to the American Steel Foundries as works manager of the Thurlow plant, Chester, Pa., which position he held until 1925.

• • •



FRANK A. MERRICK

FOLLOWING years of service with Westinghouse Electric & Mfg. Co., Frank A. Merrick, president since 1929, has been elected vice chairman of the company.

Born in Lambertville, Mr. Merrick graduated in electrical engineering at Lehigh University, in 1891. In the same year he joined the Thomson-Houston Electric Co., then served successively as consultant with the engineering firm, Blood and Hale, and as manager and chief engineer of the Steel Motor Co. When Westinghouse, in 1902, acquired the Steel Motor Co., Mr. Merrick remained.

The following year the Canadian Westinghouse Co. Ltd. was formed and he was named superintendent, later serving as manager, vice president, general manager and director. After the war he was located for

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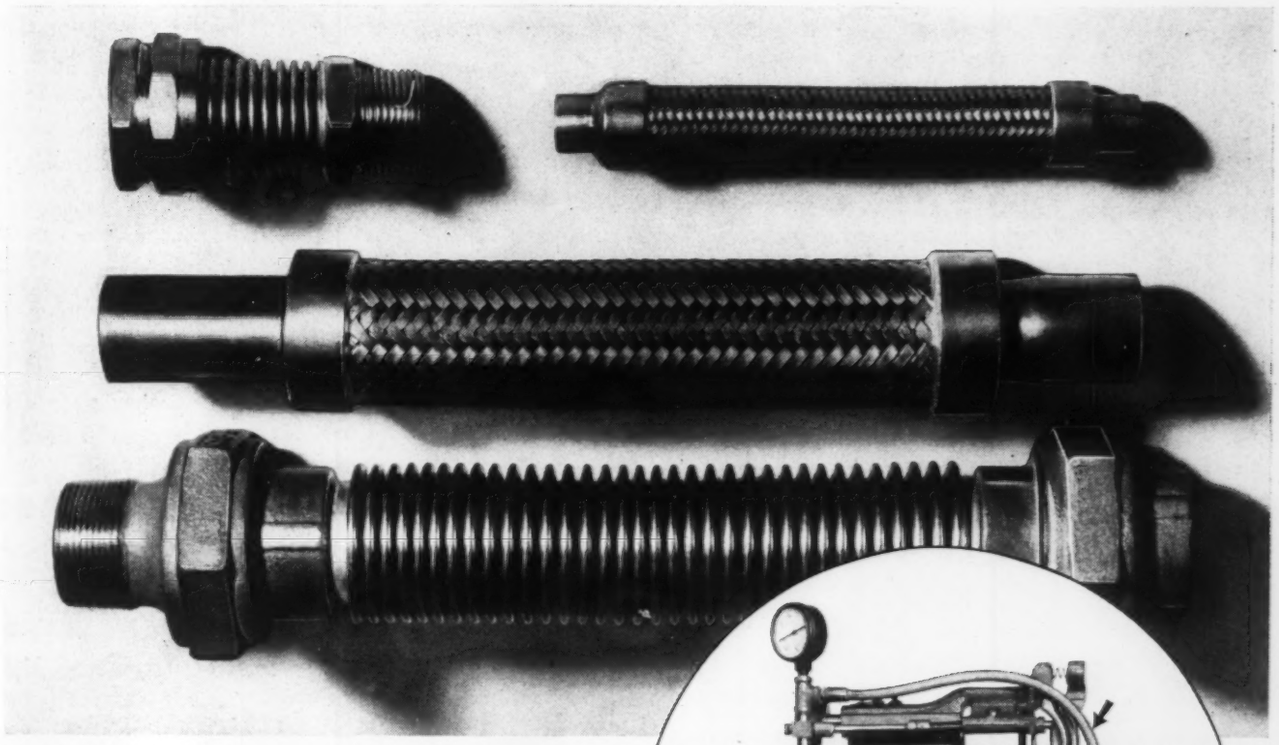

Assemblies of American Seamless Flexible Bronze Tubing with various types of end fittings used as leak-proof flexible connectors for taking up vibration, misalignment, etc.

Seamless Flexible Tubing used for conveying shoe cement on shoe making machine.

INSTALL American Seamless Flexible Bronze Tubing as original parts on your product. It's cheap insurance against the need for future servicing. Wherever you need a flexible connector or conductor... for misaligned or moving parts, for complete elimination of vibration...the only 100% safe answer is American Seamless Flexible Bronze Tubing.

For the conveyance of air, water, oil, steam or fuel, "American" has no superior. No joints, welds, laps, seams or packing are used in its manufacture. *It's seamless.* That's why it's leak-proof, and that's why more and more designers of machinery of all types are putting "American Seamless" into their specifications.

Write us about your connector problems. Our engineering department has a wealth of information on the use of "American Seamless" on all types of machinery. Consultation of this department entails no obligation.

ANACONDA

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FREE REFERENCE HANDBOOK ON SEAMLESS FLEXIBLE METAL TUBING

The most dependable *flexible* connector for conveying liquids or gases under high pressures . . . made from special high tensile strength *seamless* Bronze Tubing. Handbook contains descriptions, illustrations and valuable engineering data. Write for Bulletin SS-3.

THE AMERICAN BRASS COMPANY

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BIG NEWS

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No. 2886—Made of finest Southern Cedar, finished in blue with black tip, with degree of hardness marked in gold at top of pencil on all six sides.
17 degrees of hardness ..... **15c ea.**

No. 2888—"Chisel Point" oval shape with flat lead makes possible extremely long lines of uniform thickness. This unique construction offers the advantage of an unusually thin lead with no risk of breakage. Same wood and finish as No. 2886 with degrees of hardness marked in gold on the two sides.
In 6 hard degrees ..... **15c ea.**

No. 1018—Artist Pencil with lead sharpener at end. Made to hold No. 1904 Mars Lumograph leads. Refillable from the top. Special finger grip makes this pencil extremely easy to use ..... **50c ea.**

No. 1904—Mars Lumograph Refill Leads for No. 1018 Artist Pencil. Same as in No. 2886 Mars Lumograph Pencil. 13 degrees of hardness.
Box of 6 leads ..... **50c a box.**

Convince yourself of the merits of this unique pencil—make your own practical test. Mail coupon today.

J. S. STAEDTLER, Inc.
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MAIL COUPON TODAY

Gentlemen—Enclosed find $ ..... Please send postpaid—Mars Lumograph Pencil or leads as checked.

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two years as special representative in London, at the end of which time he returned to his duties in Canada. In 1925 he became vice president and general manager of Westinghouse Electric & Mfg. Co. at East Pittsburgh. Four years later he was elected president.

♦ ♦ ♦

HARRY M. WILLIAMS, a veteran refrigeration engineer, has resigned as manager of the commercial engineering department of Frigidaire division of General Motors Corp., to become chief engineer of National Cash Register Co. Mr. Williams is also the retiring president of the American Society of Refrigerating Engineers.

♦ ♦ ♦

DR. ROBERT E. DOHERTY, president, Carnegie Institute of Technology, Pittsburgh, has received the 1937 Lamme medal of the American Institute of Electrical Engineers for his "extension of the theory of alternating current machinery, his skill in introducing that theory into practice, and his encouragement of young men to aspire to excellence in this field."

♦ ♦ ♦

A. L. BERGSTROM has been appointed executive engineer to coordinate some of the varied engineering activities of the Timken Roller Bearing Co. This appointment was made following the resignation of ERNEST WOOLER, formerly chief engineer. R. M. RIBLET will become chief engineer of the automotive division with J. B. BAKER as his assistant.

Obituaries

BENNETT MATTINGLY BRIGMAN, dean, Speed Scientific school, University of Louisville, and vice president of the American Society of Mechanical Engineers, died of a heart ailment on Feb. 8, at the age of 56.

Dean Brigman, born at Louisville in 1881, received his B. S. and M. S. degrees from the University of Louisville. After two years as acting engineer of tests for a railway company, he began his teaching career as instructor in manual arts at the University School (preparatory). From 1907 to 1916 he was instructor in manual arts at the du Pont Manual Training School, Louisville, and during the last four years of this period he served as part-time instructor in drawing at the university of Louisville. He then became assistant professor of drawing and physics at the university, and in 1919, professor of engineering and drawing.

From 1916 to 1923, Dean Brigman engaged in private consulting work in patents and machine design, and in 1923 was called upon to organize a school of engineering at the University of Louisville, of which, in 1924, he became dean of engineering.

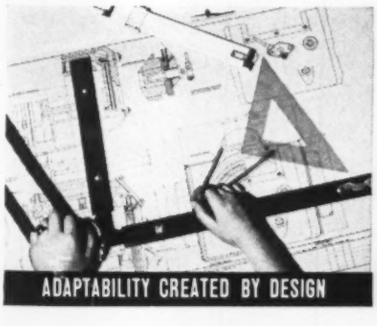
Above is shown the Hyatt Roller Bearing as applied with and without the separable inner race, which is a distinctive feature of Hyatt design.

ADAPTABILITY

FROM THE HAND OF HYATT

THERE IS INFINITE VARIETY to the applications of Hyatt Roller Bearings. From the smallest installation to the largest their inherent adaptability, simplicity of design, and strict conformance with standard dimensions help simplify and improve mechanical design wherever they are used.

Meeting all conditions of loads and speeds, these better bearings promote economical operation and prolong the useful life of the motor cars, farm machinery and industrial equipment they so universally serve. Hyatt Bearings Division, General Motors Corporation, Harrison, N. J.



ADAPTABILITY CREATED BY DESIGN

ADAPTABILITY PROVED BY TEST

HYATT ROLLER BEARINGS

PRECISION

BEARINGS

in 108 distinct series

To the machinery world, NORMA-HOFFMANN offers the most comprehensive line of anti-friction bearings in America—108 distinct series—ball, roller, needle and thrust types—1/8" to 21" bore, metric and inch sizes.

Many of these types have been pioneered by NORMA-HOFFMANN engineers to meet specific requirements growing out of advancing methods in machine design, manufacture and operation.

Today, with this wide choice of PRECISION BEARINGS available, engineers are no longer compelled to adapt their designs to the comparatively few standard bearing types of past years. There's a PRECISION BEARING for every load, speed and duty.

Write for the Catalog. Let our engineers work with you.

NORMA-HOFFMANN BEARINGS CORP'N.

STAMFORD, CONN., U. S. A.

"NORMA-HOFFMANN"
PRECISION BEARINGS
BALL, ROLLER AND THRUST

Patents Relating to Drives and Controls

AN improved type of extremely small electric motor, invented by the late Col. Jacob Schick of Stamford, Conn., is depicted in *Fig.* 1 as it appears when built into a dry shaver. This interesting development is covered by patent No. 2,038,405.

This motor, which operates on either direct or alternating current, is designed to be of minimum size and weight. Starting and stopping is effected without recourse to a switch, by means of a starting wheel screwed on the armature shaft, the knurled rim of this wheel being exposed through a recess in the casing. The motor can be speeded up or slowed down by means of a lever extending to the outside of the case. This lever — in a manner made clear by the drawing — varies the distance between the contact points and cam and simultaneously varies the distance between the cam and the pivot of the contact arms.

The motor is of magnetic type, comprising a U-shaped field magnet of laminated iron and an unwound armature. The cross arm of the field magnet is surrounded by a coil which is in series with the contacts of the make-and-break device, the arms of which are of insulating material. The above-mentioned cam which actuates this make-and-break device is simply the flattened end of the armature shaft.

In the shaver depicted herewith power is transmitted from the motor to cutting head by means of

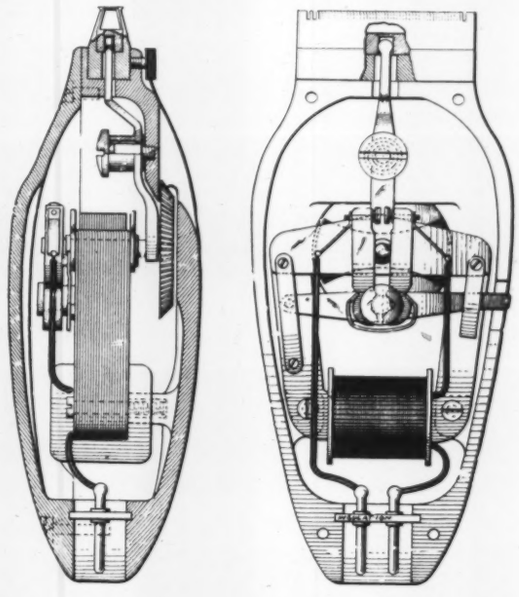


Fig. 1—*Powerful 4-ounce motor can be built into compact case which fits the hand*

G-E APPLIANCE WIRING AND CONTROL DEVICES

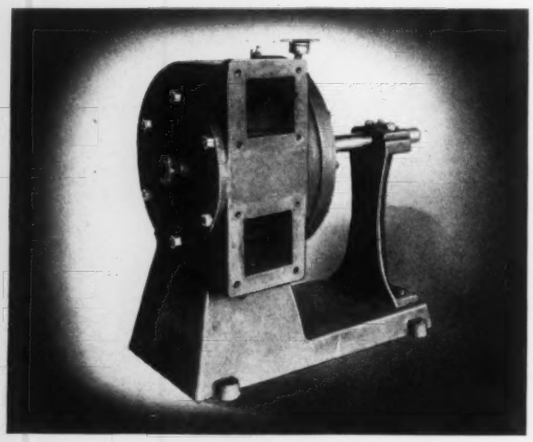
Appliance wiring and control devices — important to appliance design — are always available in General Electric's popular line of Accessory Equipment.

These devices can be used to advantage on your appliances, to improve design and performance, and to reduce assembly and service costs. They will give your appliances important sales features favored by customers.

The General Electric Accessory Equipment Section is equipped to serve appliance manufacturers. Take advantage of the wide variety of G-E appliance wiring and control devices, or consult G-E accessory engineers if your product requires specialized attention. Custom service is always at your disposal.

For further information, write to Section Q-8344, Appliance and Merchandise Department, General Electric Company, Bridgeport, Connecticut.

GENERAL ELECTRIC



Over 600,000 Ago!

This is the original Viking Pump. Although pensioned away today in a corner of the Viking Plant it is still mechanically good . . . still capable of a full day's work. Built in 1907, it has served as a splendid forerunner to the more than 600,000 Viking Rotary Pumps which have followed.

Experience gained in these many years of building better pumps has taught Viking Engineers how to effectively meet the cooling liquid pump problems on your particular machines. Check, carefully, Viking's bulletin on Coolant Pumps for Machine Tools.

VIKING PUMP CO.
CEDAR FALLS, IOWA.

"Using CHACE Thermostatic BIMETAL in our FRASER FURNACES"

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Fraser Floor Furnaces are compact, gas burning, heating units. Occupy very limited space and produce heat at low cost. Chace Bimetal is the active element used in the safety pilot, which automatically shuts off the gas supply in event pilot is extinguished.

For dependable automatic movement in your product investigate the possibilities of Chace Precision Thermostatic Bimetals. Sold in sheets, in strips and in finished form ready for assembling in your product.

CHACE THERMOSTATIC BIMETAL

W. M. CHACE CO.
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a center-pivoted lever. The forked lower end of this lever straddles an eccentric on the armature shaft while its rounded upper end engages a slot in the sliding inner member of the cutting head which thus is oscillated at high speed as the armature whirls.

The design of this motor is such that it not only is surprisingly powerful for its size and of extremely long operating life, but also is of such a nature that it fits neatly into a compact casing designed to fit the hand. The motor weighs about four ounces.

Clutch Is Adapted for Presses

SHOWN by *Fig.* 2 is a magnetic clutch invented by Howard E. Hodgson of Wauwatosa, Wis. His patent—No. 2,102,318—is assigned to Cutler-Hammer Inc., Milwaukee. Driving parts are designed for large flywheel effect in proportion to weight—the driven element for low inertia.

The flywheel member carries the magnetic element. This embodies two friction rings between which is

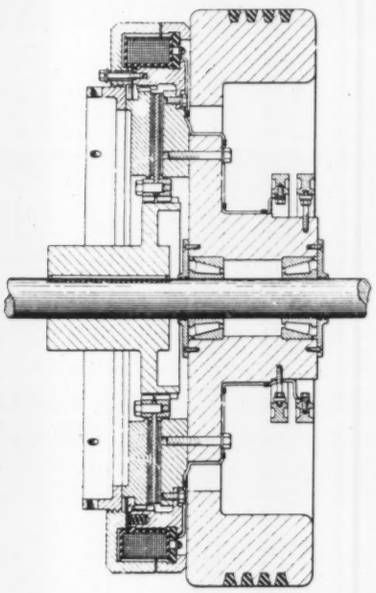
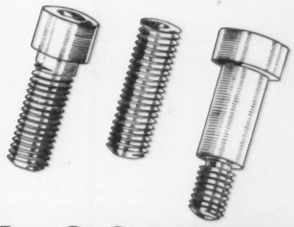


Fig. 2 — Magnetic clutch is designed to have high inertia in drive elements but low inertia driven elements, making it especially adaptable to power press control

"sandwiched" a third, slidably mounted on multiple splines on a hub keyed to the shaft. The annular magnet comprises field and armature with windings on an insulated bobbin on field member. Field and armature have one set of annular pole projections surrounding the winding and another set within the winding, the first having cooperating pole faces adjacent to right end of winding and last set with pole faces at left.

Armature slides on bushings screwed to field member, normally being held away from it by springs. Left hand friction ring, slidable on multiple splines in pole projection, also has an annular projection abutting thrust ring on armature. Magnetic attraction of armature moves outer ring to right, squeezing center ring thus establishing drive from flywheel.

From start *to finish* 

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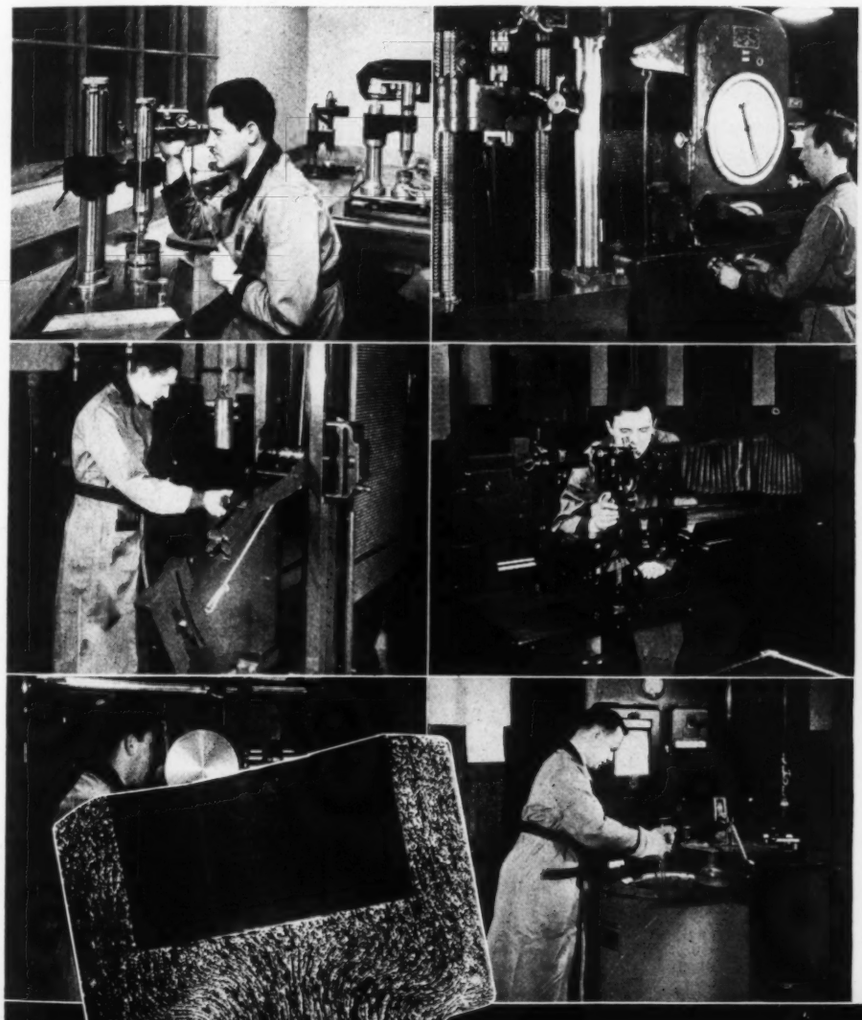

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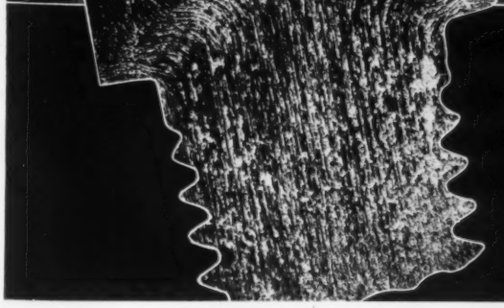

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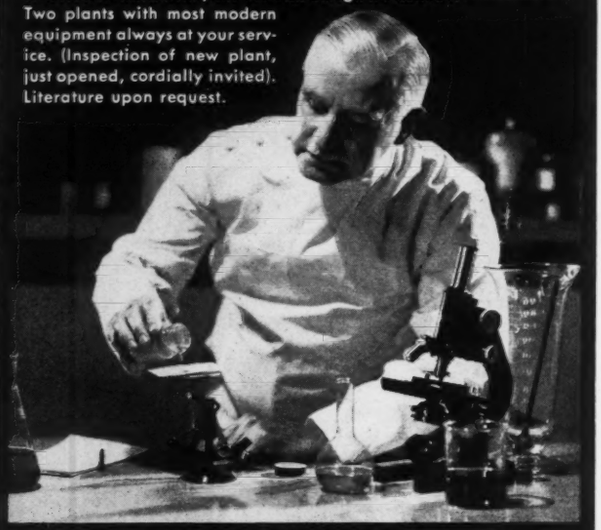

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NEW Materials and Parts

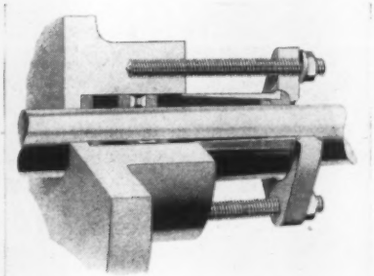
Motor Driven Time Control Unit

PRIMARILY designed for installation as an integral part of various electrical appliances and machines operative on a predetermined time basis, such as processing machinery, household appliances, diagnostic and therapeutic apparatus, a synchronous motor-driven timer (model A) has been placed on the market by R. W. Cramer & Co. Inc., 68 Irving Place, New York. Synchronous motor is slow speed, self-starting with enclosed gear train operating in oil. Torque is several times greater than required to throw switch. Timing scales range from a one-minute dial with divisions in seconds up to an eight-hour dial with 15-minute divisions.

Timer motor operates gear train enclosed in oil filled case. Torque is sufficient to throw switch under any conditions

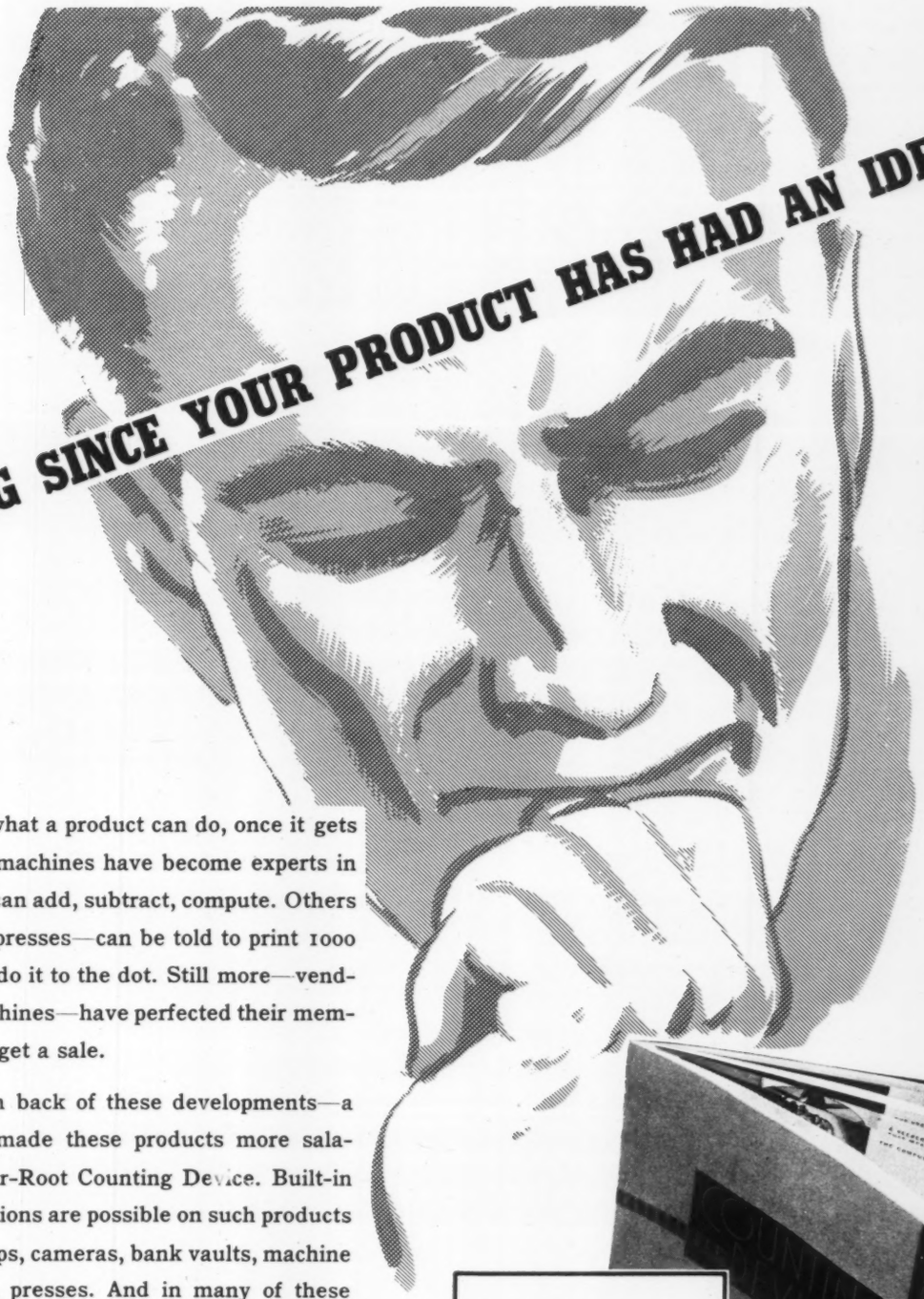
Packing Gland Is Self-Aligning

NEW type, conical, self-aligning packing gland adapted for ammonia, air and gas compressors, reciprocating pumps and engines has been announced by Crane Packing Co., 1800 Cuyler avenue, Chicago. The gland cup is made from nickel steel and fine bearing bronze, machine finished inside and out. The inner surface of the cup is tapered, the effect of which is to



Gland cup is made from nickel steel and fine bearing bronze. Inner surface of cup is tapered

HOW LONG SINCE YOUR PRODUCT HAS HAD AN IDEA?



It's surprising what a product can do, once it gets an idea. Some machines have become experts in mathematics—can add, subtract, compute. Others —like printing presses—can be told to print 1000 sheets and will do it to the dot. Still more—vending and slot machines—have perfected their memories. Never forget a sale.

And one idea in back of these developments—a push that has made these products more salable—is a Veeder-Root Counting Device. Built-in Counter applications are possible on such products as—looms, pumps, cameras, bank vaults, machine guns, elevators, presses. And in many of these products the application of the Counting Device has been new and entirely different.

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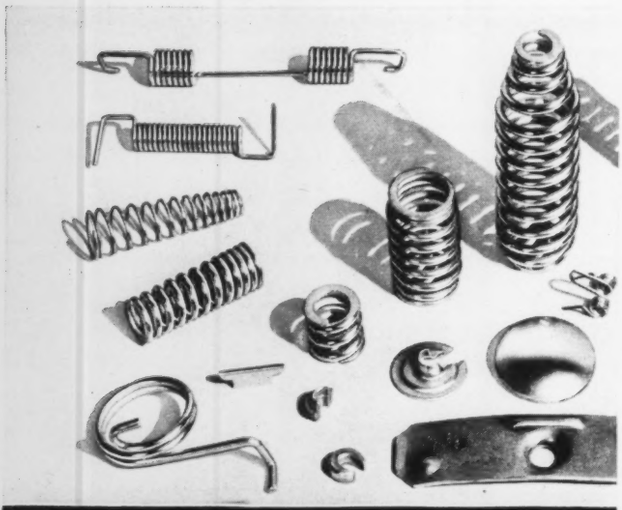
66 It is a Raymond policy to investigate new materials and methods which appear to promise improved spring performance 99

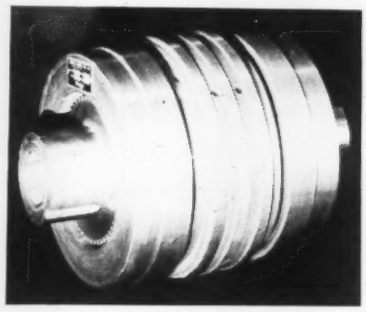


RAYMOND SPRINGS
RAYMOND MFG. CO.
DIVISION OF ASSOCIATED SPRING CORPORATION
280 So. Centre St. - - - CORRY, PA.

force the packing around the rod with each stroke of the piston. Ball and socket construction compensates for "float" present in varying degrees in pumps and compressors and prevents wedging or distortion of the cup where it meets the cast steel follower if the studs are drawn up unequally when the gland is being adjusted. This packing gland is adaptable for use on both the steam and liquid ends of pumps and the power and compression ends of gas compressors.

Magnetic Clutch Announced

UTILIZING the principle of two clutches energized with a single magnet, a high duty (style ED) duplex magnetic clutch has just been released by Stearns Magnetic Mfg. Co., Milwaukee. Feature of

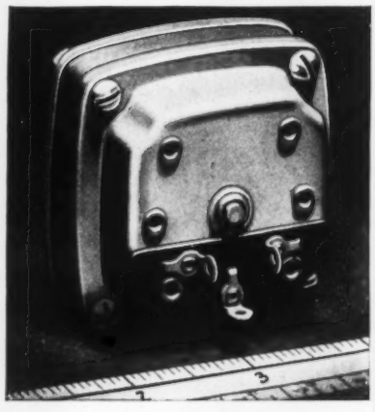


Clutch may be rotated in either direction and powerful magnetic field makes it applicable for severe operation

this clutch is that it may be rotated in either direction, giving particular advantages in many applications. Its extremely simple design, few moving parts and powerful magnetic field make it adaptable for a wide variety of machine applications.

Motor Developed for Radio Tuning

DESIGNED for radio pushbutton tuning, a tiny motor (model R), equipped with mounting and gear assemblies has been placed on the market by



Ample heat radiating area is provided by housing of tiny motor, and thermostatic control protects against accidental burning out

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"General Considerations in Designing Mechanical Springs"

BY A. M. WAHL

This series of technical articles, originally published by MACHINE DESIGN in 1930, reprinted in 1931 and completely revised and presented in four parts this year, is now available in reprint form. For those engineers and designers preferring this information in convenient form, copies of the reprint may be secured for 50 cents prepaid from the Service Department, MACHINE DESIGN, Penton Building, Cleveland.

This work will prove of inestimable assistance to every designer of machinery. Information is provided that will enable preliminary calculations to be made on design of all types of springs.

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Compression and Tension Springs.
Torsion and Spiral Springs.

Formulas for design and complete information on spring application make this series a valuable addition to handbook and engineering data files. Its use will reduce the number of spring failures due to "guess-work". When engineers use Dr. Wahl's findings in specifying springs, manufacturers will be in a better position to supply exactly the required spring with assurance.

A. M. Wahl, Research Engineer, Westinghouse Research Laboratories, in addition to being author of "General Considerations in Designing Mechanical Springs" and other articles on the same subject, has prepared numerous technical articles on pipe bends, stress analysis of machine parts, stress concentration and photoelasticity. He is co-author of a book on "Plasticity" and collaborated in revising the "Machine Elements" section of Marks' Mechanical Engineers Handbook, 1930 Edition. Dr. Wahl received his B. S. degree from Iowa State College and his PhD degree from the University of Pittsburgh. He has been a member of the Westinghouse Research staff since completion of his engineering and design school training in 1926, specializing in research work on mechanical and electrical engineering problems arising in connection with the design of electrical machinery. He is a member of the American Society of Mechanical Engineers.

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by bronze ring pressing against bearing race or shaft shoulder.
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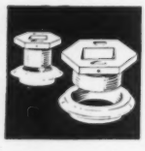

Forged and machined fittings for steel barrels, drums, tanks and metal packages, including plugs, rings and flanges. Very rugged for use in transportation of alcohol, turpentine, oils and other expensive liquids.

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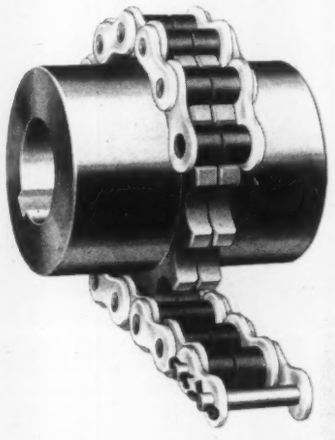
ample heat radiating area plus thermostatic protection against accidental burning out. Overall dimensions are 2⅛ inches square by 1⅝ inches.

Improved Coupling Developed

CONSISTING of two cut tooth sprocket wheels connected by a piece of specially constructed single-width steel roller chain, a flexible coupling (type RCB) has been announced by Link-Belt Co., 519 N. Holmes avenue, Indianapolis. The chain has

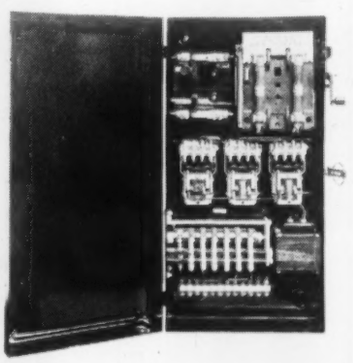


Coupling connecting chain has divided rollers which provide independent roller action for each coupling sprocket

a divided roller feature which combines the advantage of double roller chain with the more rugged and simple construction of single-width chain. The divided roller provides independent roller action for each sprocket and as the contact between roller and sprocket causes the roller to revolve on its bushing, any tendency to scuff the rollers and sprocket teeth is avoided. When specified a grease-retaining housing is provided for the coupling.

Control for Air-Operated Machines

DEVELOPED primarily to minimize the possibility of false operation, a control system for punch presses actuated by air clutches has been designed by

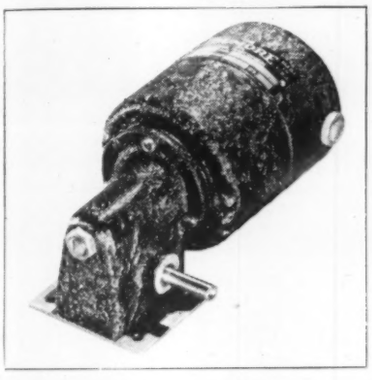


Ground at any point in control system is impossible as all contacts are connected in secondary circuit of a special transformer

General Electric Co., Schenectady, N. Y. The new equipment, consisting of a magnetic control panel, a rotating cam-type limit switch and oilproof push-button stations, permits four different operations from "continuous stroke" to "inching," while the safety hazard of false operation arising from wiring failures caused by oil and severe vibration has been practically eliminated. All pushbuttons, limit switch contacts, and contactor coils are connected in the secondary circuit of a specially designed control transformer. This makes impossible a ground at any point in the system to energize the solenoid falsely.

Fractional Horsepower Gearmotor

FRACTIONAL horsepower universal motor with gear reduction unit for such applications as coin operated machines, aircraft cowl flaps, advertising



Shaft of gear reduction unit extends from both sides giving either direction of rotation without reversing motor armature

machines, fllm rewinders, etc. has been introduced by The Dumore Co., Racine, Wis. The types HL and HVL motors—the types H and HV with single reduction gear units—are identical except the former has a totally enclosed housing and 30-minute rating; the latter, a ventilated housing and continuous rating. Mounting on the base of the gear unit provides rigidity. A double gear-shaft extension makes possible either direction of rotation without changing the rotating of the armature. Motors are available operating on either 115-volt direct or alternating current and in five gear ratios. Complete weight is two pounds.

Air Circulator Motor Is Streamlined

AIR circulator motor, especially shaped to blend with the usual air circulator design and to offer the least possible resistance to the flow of air has been placed on the market by The Ohio Electric Mfg. Co., 5917 Maurice avenue, Cleveland. The motor is designed with barrel frame to facilitate mounting in a ring or yoke. Split-phase windings for 1150 of 1725 RPM with start and stop pull switch are provided, and also windings for speeds of 860 and 1140 and 1725

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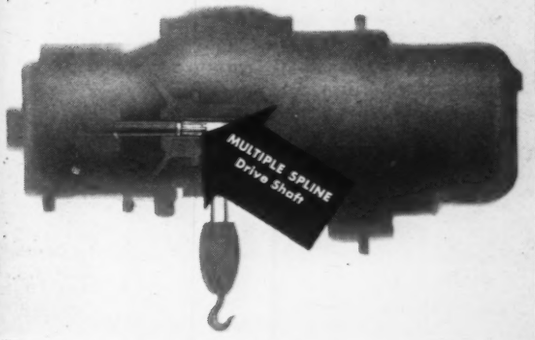
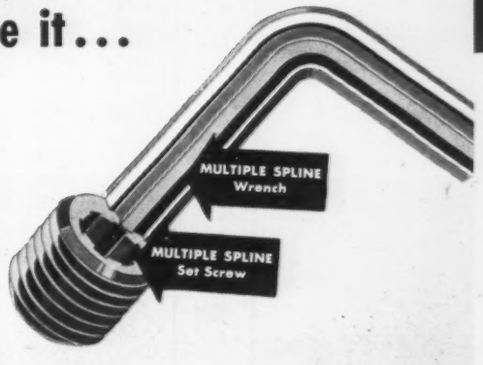


BRISTO SOCKET SCREWS use it...



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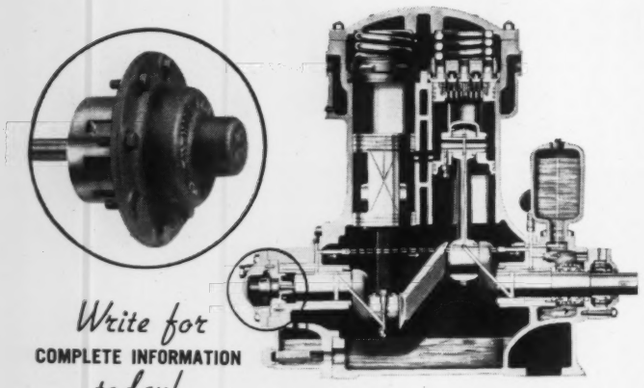

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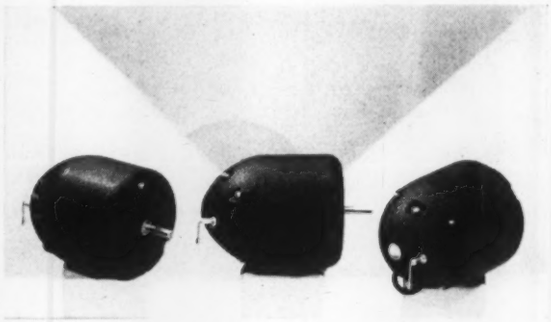
RPM. With separate transformer capacitor control the motors will operate at three different speeds. Motors are available in 1/8, 1/6 and 1/4 horsepower sizes to operate fans from 20 to 32 inches in diameter. Finish may be crinkle or smooth enamel.



By use of special windings two speeds are possible with motor, and transformer control gives range of three speeds.

Meetings and Expositions

April 25-27—

American Gear Manufacturers' association. Annual meeting to be held at General Brock hotel, Niagara Falls, Canada. J. C. McQuiston, Penn Lincoln hotel, Wilkinsburg, Pa., is secretary. Meeting dates were recently changed from earlier in April to the above.

May 14-19—

American Foundrymen's association. Forty-second annual convention to be held at the Public Auditorium, Cleveland. Dan M. Avey, 222 West Adams street, Chicago, is secretary-treasurer.

June 20-24—

American Institute of Electrical Engineers. National annual summer convention to be held at Mayflower hotel, Washington. H. H. Henline, 33 West Thirty-ninth street, New York, is national secretary.

June 27-30—

American Society of Agricultural Engineers. Annual meeting to be held at Asilomar hotel, Pacific Grove, Calif. Raymond Olney, St. Joseph, Mich., is secretary.

June 27-July 1—

American Society for Testing Materials. Annual meeting to be held at Chalfonte-Haddon hall, Atlantic City, N. J. C. L. Warwick, 260 South Broad street, Philadelphia, is secretary.

Mechanical Spring Design

(*Continued from Page* 30)

done in Part II. Thus assuming that the yield point s_y (in tension) and the endurance limit s_e' for pulsating *bending* stress (0 to maximum) are known, then Eq. 14 of Part II may be used directly, using instead of K_c, the factors K_1 or K_2[9]. It so happens that K_1 or K_2 differ but little from the values for K_c as given by *Fig.* 4 of Part II, and hence the charts of *Figs.* 9 to 11 of Part II may be used, the limitations of these charts as mentioned there being kept in mind.

TENSION SPRINGS—The curves of *Fig.* 6 may also be used to estimate the stress concentration near the loop ends of helical tension springs, i. e. at point *a* (*Fig.* 7). It is known that tension springs under fatigue loading usually fail at such points and hence in practice it is customary to use a stress figure for tension springs about 25 per cent less than that which would be used for compression springs under similar service. Referring to *Fig.* 7, let r_1 be the smallest radius of curvature of the bend in the loop at point *a*. Then taking $c = 2r_1/d$ the stress concentration factor K_1 may be obtained from *Fig.* 6. Since in many cases the radius r_1 is quite small in practical springs, the factor K_1 may reach quite high values. This means that a definite weakening effect is present particularly under fatigue loading. It should be noted that at point *a* (*Fig.* 7) a combination of bending and torsion stresses exists, the value of which depends on the exact shape of the loop. When these stresses are combined on the basis of the *maximum shear theory* to give an equivalent stress which in turn is multiplied by the factor K_1 (as determined from *Fig.* 6 using the known value of $2r_1/d$) the resultant stress may reach high values, particularly for small values of r_1. The importance of making the minimum loop radius r_1 (*Fig.* 7) as large as possible in actual tension springs, especially those subject to fatigue, is thus clear.

SUMMARY—Summing up the work covered in this series of articles, in Part I some general considerations involved in the design of mechanical springs were discussed, particularly from the viewpoint of the determination of working stress and the choice of the factor of safety. In Part II, charts were given for facilitating the calculation of stress and deflection in helical compression springs.

Values of working stress used in practice and additional endurance test results on springs made from different steels and of different sizes were discussed in Part III; in this part formulas for the calculation of buckling loads in helical compression springs were also given. In the final article more accurate formulas were suggested for designing square and rectangular wire compression springs, and torsion springs.

9. It is assumed that s_{min} is not negative (i.e. that the direction of the moment M is not reversed). For cases where this is the case, Eq. 13 of Part II may be used, taking s_e as the endurance limit for reversed *bending* stress.

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MANUFACTURERS' PUBLICATIONS

BEARINGS—Complete sleeve-type bearing information and data are covered in a 64-page catalog No. 380, issued by Johnson Bronze Co., New Castle, Pa.

BEARINGS—Ten features of Hyatt bearings are discussed in an attractively prepared booklet of Hyatt Bearings Division, General Motors Corp., Detroit.

CONTROLS (ELECTRICAL)—Type HD heavy duty pushbuttons is the subject of a new publication describing a line of pushbuttons operating on alternating or direct current pilot circuits issued by Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

CONTROLS (ELECTRICAL) — Relays, timers and thermostats are all thoroughly treated in a technical catalog just issued by Struthers-Dunn, Inc., 139 N. Juniper street, Philadelphia.

ELECTRICAL EQUIPMENT—Attractively prepared, several bulletins on new developments of the General Electric Co., Schenectady, N. Y., have been released. These cover photoelectric relays (GEA-2516), motors for belt-driven blowers and air conditioners (GEA-2627), control equipment for traveling electric cranes (GEA-2545), multispeed induction motors (GEA-1884A), push-button stations (GEA-2473) and combination magnetic switches (GEA-1587C).

ENGINEERING DEPARTMENT EQUIPMENT — Booklet on graph sheets and co-ordinate papers, illustrating the use of typical graph forms in a concise manner, has been released by Keuffel & Esser Co., Hoboken, N. J.

FITTINGS (PIPE)—Unique types of self-contained pipe joints that connect pipe without threading are contained in a 16-page illustrated booklet of S. R. Dresser Manufacturing Co., Bradford, Pa.

MOTORS—Bulletin 251 of Crocker-Wheeler Electric Mfg. Co., Ampere, N. J., describes a new alternating current, three-phase, polyspeed motor developed by the company. Efficient and stable regulation over a wide range is possible with the motor.

MOTORS—Fractional horsepower motors of all types and windings and a general description of the Bodine motor line are described in an 8-page bulletin, 1020-A, prepared by Bodine Electric Co., 2264, W. Ohio street, Chicago.

PNEUMATIC EQUIPMENT—Catalog 226 of Hanna Engineering Works, 1765 Elston avenue, Chicago, covers the line of air cylinders, hoists and valves made by the company. Capacities, dimensions, etc., are included.

SPEED REDUCERS—A wide variety of speed reducer models is described, and specifications given, in a small booklet of Charles Bond Co., 617 Arch street, Philadelphia.

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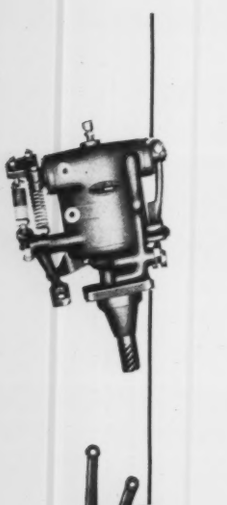

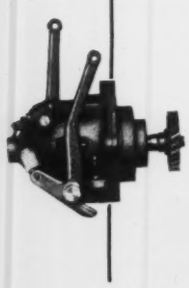

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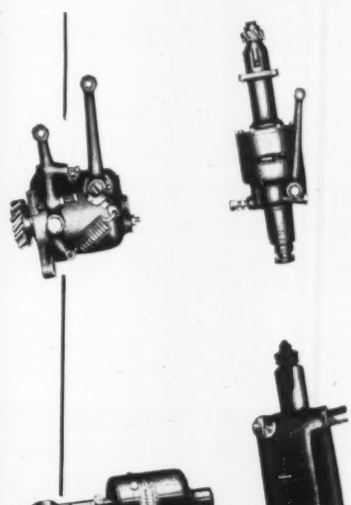

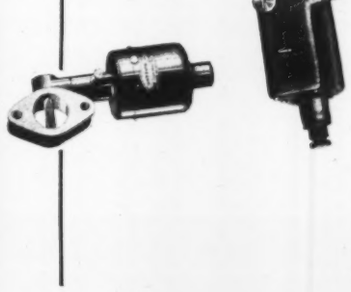

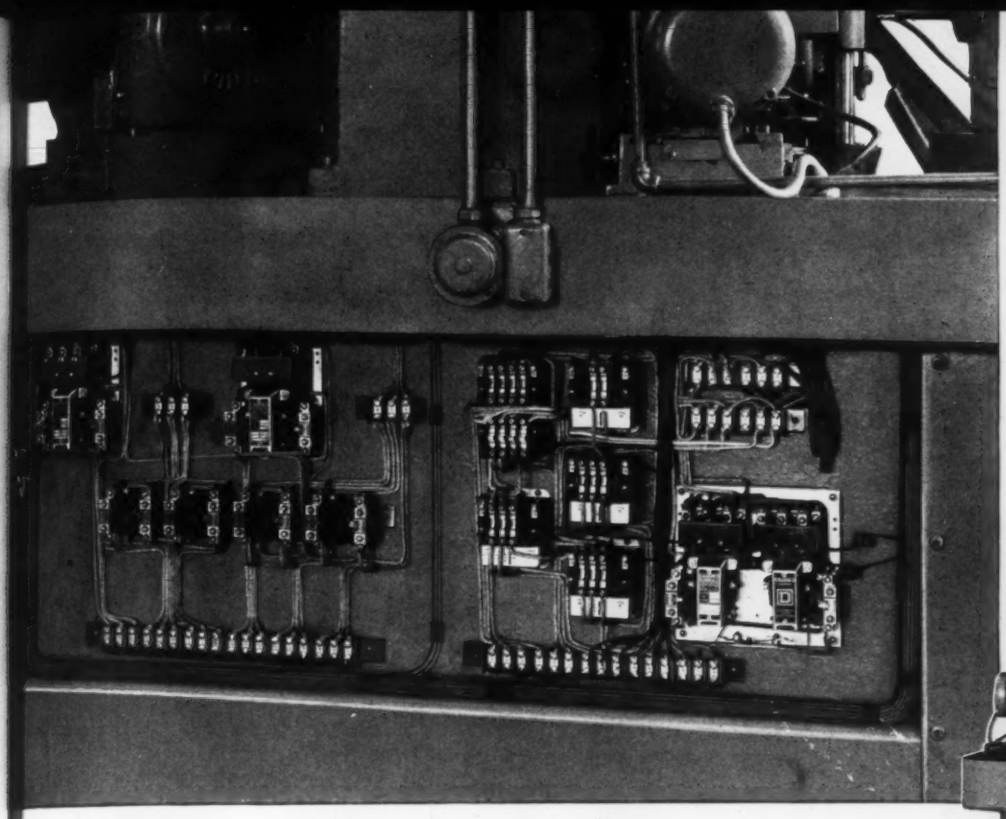

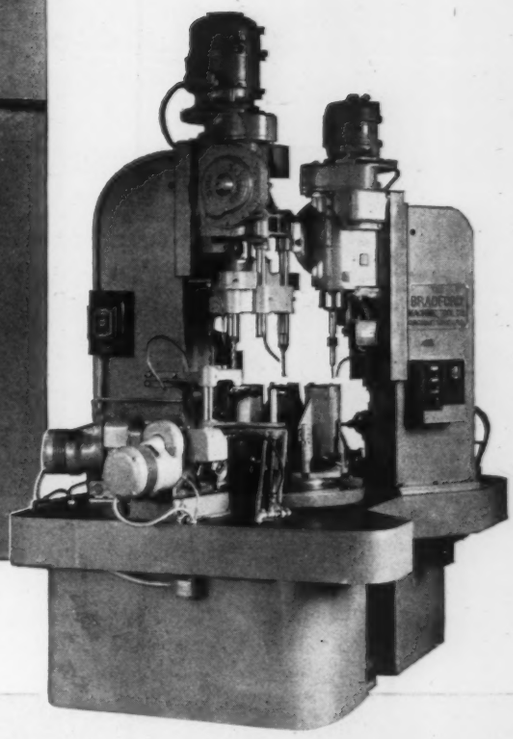

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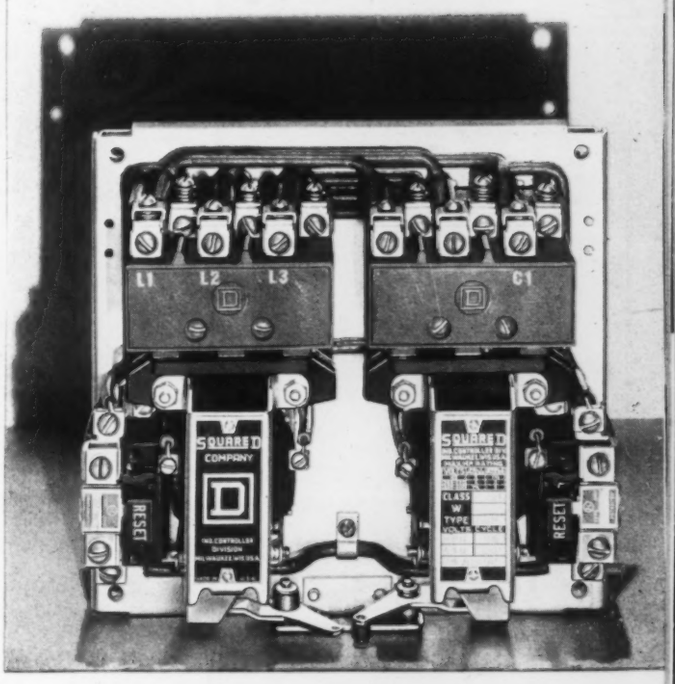

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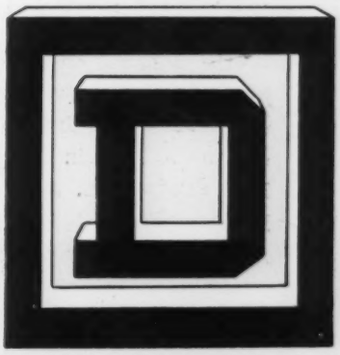

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